NASA CR- 66156

DEVELOPMENT OF A PROPORTIONAL TWO STAGE PRESSURE FEEDBACK PNEUMATIC VALVE FOR 2000°F SOLID PROPELLANT SYSTEMS

Prepared Under Contract No. NAS 1-4102
Phase III By
Research and Development Department
Vickers Incorporated
Division of Sperry Rand Corporation
Troy, Michigan 48084

FOR
LANGLEY RESEARCH CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JULY 1966

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ABSTRACT

The objective of Phase III of NASA Contract NAS 1-4102 was to conduct research and development efforts to accomplish the design and development of a flightweight prototype proportional two-stage pressure feedback valve for 2000°F solid propellant SITVC systems.

Development tasks included the application of secondary flow amplification for control of high primary pneumatic mass flows independent of extraneous internal valve flow and mechanical forces. Incorporation of closed loop pressure feedback logic was developed to achieve functional static accuracy and dynamic response. Six (6) flightweight prototype models were fabricated and experimental development tests were conducted under both gaseous nitrogen and 2000°F pneumatic supply environments.

The analog analytical study provided design parameters to optimize the dynamic response characteristics and flightweight design criteria. Analog investigations included determining feasibility and application of scaled mass flow, system pressure levels, and capability potential for 5500°F solid propellant systems. Materials and design stress investigations were included for 5500°F systems.

The analytical study and experimental development investigations showed that the concept of a flightweight proportional two-stage

valve incorporating closed loop pressure feedback logic for high temperature solid propellant pneumatic systems is practical, since all functional static and dynamic operating characteristics were achieved.

FOREWORD

Vickers Incorporated, Research and Development has completed the research, design, and development of a flightweight prototype proportional two-stage pneumatic valve with closed loop pressure feedback applicable for 2000°F solid propellant pneumatic systems. This work was completed under Phase III of NASA Contract NAS 1-4102 entitled, "Secondary Injection Thrust Vector Control For High Altitude Nozzles."

Phase III of the subject contract was initiated on July 6, 1965 and completed on May 20, 1966. The contract was performed under the technical cognizance of Mr. John Riebe of the Langley Research Center, Hampton, Virginia.

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NOMENCLATURE

Symbol	Description	Units
A	Orifice Effective Area	$inch^2$
A_1	Upstream Servo Orifice Area	$inch^2$
A_3	Downstream Servo Orifice Area	$inch^2$
A_b	Feedback Bellows Effective Area	$inch^2$
A_c	Effective Area-Spool-Servo Control	$inch^2$
A_d	Effective Area-Spool-Pneumatic Rate	$inch^2$
A_ℓ	Effective Area-Injection Load Orifice	$inch^2$
A_m	Servo Metering Area	$inch^2$
C	Constant	--
C_1	Orifice Flow Coefficient (Sonic)	$\sqrt{°R}/sec.$
C_d	Orifice Discharge Coefficient	--
D_3	Diameter of Downstream Servo Orifice	inch
e	Servo Error Signal	lb/inch
F_a	Output Thrust "A" Side	lbs
F_b	Output Thrust "B" Side	lbs
F_s	Friction-Stiction Force	lbs
G	Acceleration Constant (32.2)	ft/sec^2
I	Electrical Current	milliamps
K	Ratio of Specific Heats	--
K_a	Pneumatic Spring Rate	lb/inch
K_r	Torque Motor Spring Rate	lb/inch/rad
M	Molecular Weight	lbs
$\dot{M}$	Mass of Main Spool	$lb/sec^2/ft$

Symbol	Description	Units
m	Denotes Maximum	--
M_A	Electrical Current	milliamps
ΔP	Pressure Differential	psi
P_A	"A" Side Discharge Pressure	psi
P_B	"B" Side Discharge Pressure	psi
P_c	Servo Control Pressure	psi
P_d	Discharge Pressure	psi
P_g	Gas Generator Pressure	psi
P_h	Internal Case Pressure	psi
P_{ca}	Servo Control Pressure "A" Side	psi
P_{cb}	Servo Control Pressure "B" Side	psi
P_u	Upstream Absolute Pressure	psia
R	Gas Constant	ft/°R
r	Torque Arm Radius-Servo Jet	inch
r_2	Torque Arm Radius-Feedback Bellows	inch
S	LaPlace Transform	1/sec.
T	Main Spool Stroke	inch
T_f	Feedback Torque	lb/inch
T_g	Generator Gas Temperature	°F
T_i	Torque Input	lb/inch
T_o	Output Gas Temperature	°F
T_s	Servo Torque Feedback	lb/inch
T_u	Upstream Stagnation Temperature	°R
t	Time	seconds
V_A	Discharge Manifold Volume "A" Side	$inch^3$

Symbol	Description	Units
V_B	Discharge Manifold Volume "B" Side	$inch^3$
V_c	Servo Control Spool Volume	$inch^3$
$\dot{\omega}$	Pneumatic Flow Rate	lb/sec
$\dot{\omega}_A$	Output Mass Flow Rate "A" Side	lb/sec
$\dot{\omega}_B$	Output Mass Flow Rate "B" Side	lb/sec
$\dot{\omega}_g$	Generator Mass Flow Rate	lb/sec
$\dot{\omega}_s$	Servo Mass Flow Rate	lb/sec
X	Servo Yoke Stroke or Gap	inch
α	Servo Metering Orifice Area/Upstream Orifice Area Ratio	--
θ_1	Servo Angular Displacement	radians
θ_2	Spool Position	inches
δ	Damping Coefficient	inch/lb/$\frac{sec}{rad}$

SUMMARY

This program has encompassed the research and development efforts required to accomplish mathematical analysis, design, and test evaluation of a flightweight prototype two-stage 2000°F pneumatic valve incorporating pressure feedback. The valve is intended for application to solid propellant systems, for either secondary injection thrust vector control or pitch-yaw-roll attitude control. The single-axis injection system tested, successfully demonstrated the operational performance of the flightweight two-stage on a system basis. Closed loop system performance was achieved by injection differential pneumatic feedback.

Pneumatic test firings at 2000°F were conducted in a simulated SITVC system control mode. The gas generator provided a 2000 psi pneumatic output flow of 0.704 lb/sec. for a duration of 43 seconds. The system was comprised of a solid propellant gas generator, two-stage valve, associated manifolds and injection nozzles, and the pressure feedback logic. This system successfully demonstrated proportional pneumatic flow modulation in excess of 110 to 1. The system dynamic response was evaluated with sinusoidal, ramp, and transient inputs and demonstrated a dynamic frequency response capability of 45 cps at -3 db attenuation. The valve's weight is 4.24 pounds and it requires an electrical power of 5.5 watts.

An analog computer study was conducted to select the two-stage valve parameters and also included investigations to determine the effect of major parameter variations on performance. The system concept was analytically explored over a mass flow capacity range of 0.58 to 5.0 lb/sec. at 1500, 2000, and 2700 psi supply pressures, and at 2000°F and 5500°F (aluminized) solid propellant pneumatic sources. The results of this study have indicated that the two-stage valve design concept has a potential of being uprated to a 5500°F conceptual design including the pneumatic pressure feedback logic. This study was concluded with a materials and stress analysis.

Development of the flightweight 5500°F two-stage valve concept appears attractive for direct-engine-bleed secondary injection thrust vector control systems.

SECTION 1

INTRODUCTION

This report describes the research and development program to analyze, design, and evaluate a flightweight prototype two-stage high temperature (2000°F) pneumatic pressure feedback valve. The work has been accomplished using a 2000°F solid propellant generator which provided a flow rate of 0.704 lb/sec. An analog computer study was conducted to select the valve parameters and also included determination of the effect of major parameter variations such as mass flow level, 5500°F temperature and system pressure levels on valve performance. The ultimate objective of this program was to evaluate a flightweight configuration two-stage valve for 2000°F secondary injection thrust vector control systems and for PYR attitude control systems, which offered a valve concept potential for 5500°F solid propellant SITVC systems.

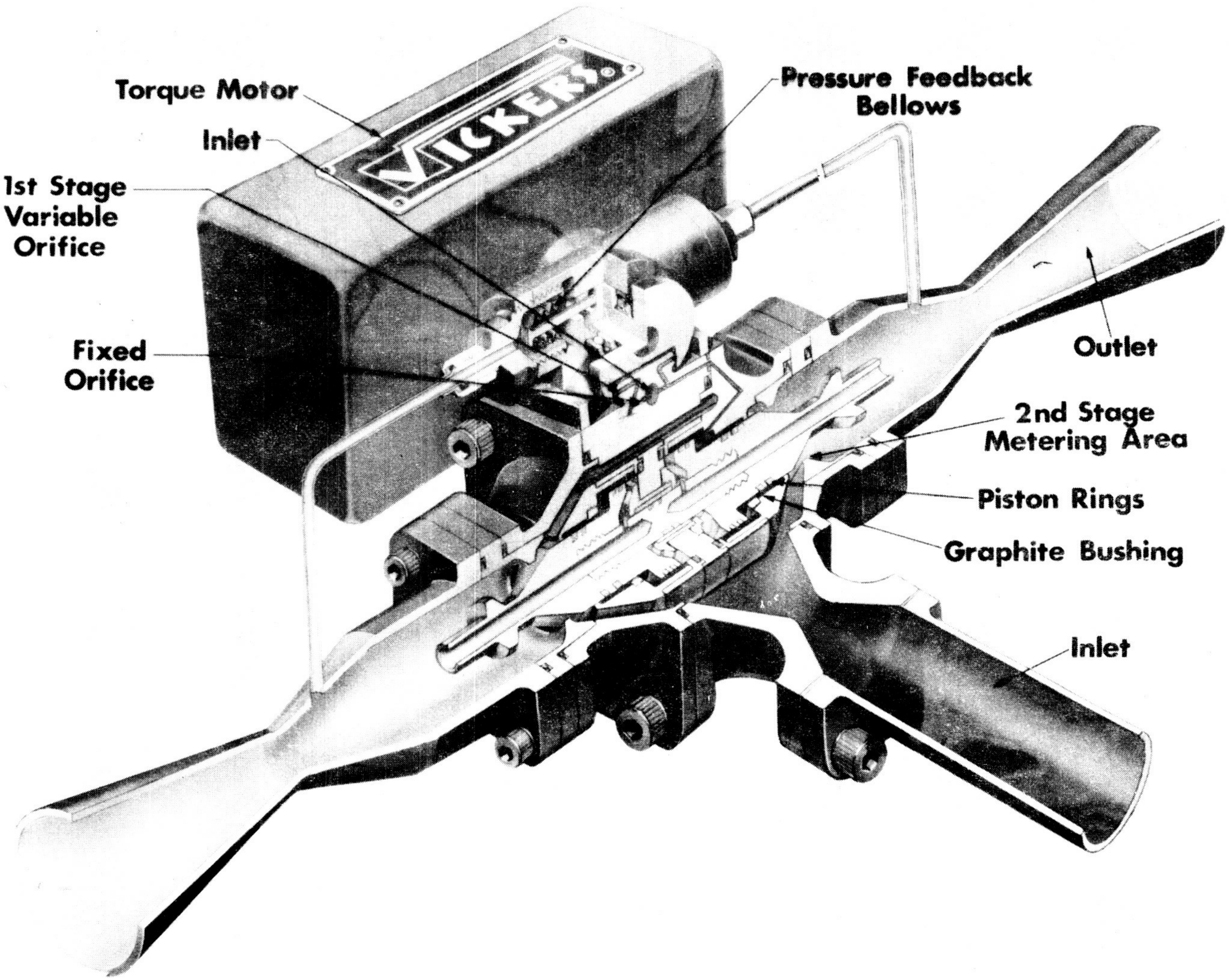
SECTION 2

PRESSURE FEEDBACK PROPORTIONAL TWO-STAGE VALVE CONCEPT

The conceptual two-stage valve design is an electro-pneumatic-mechanical control which provides a proportional output differential pneumatic pressure with respect to an electrical differential current input signal. The valve system loop is closed by incorporation of output differential pressure negative feedback. The first-stage is an open center design pneumatic amplification servo. A conceptual flight quality design of the two-stage valve is shown in Figure 1.

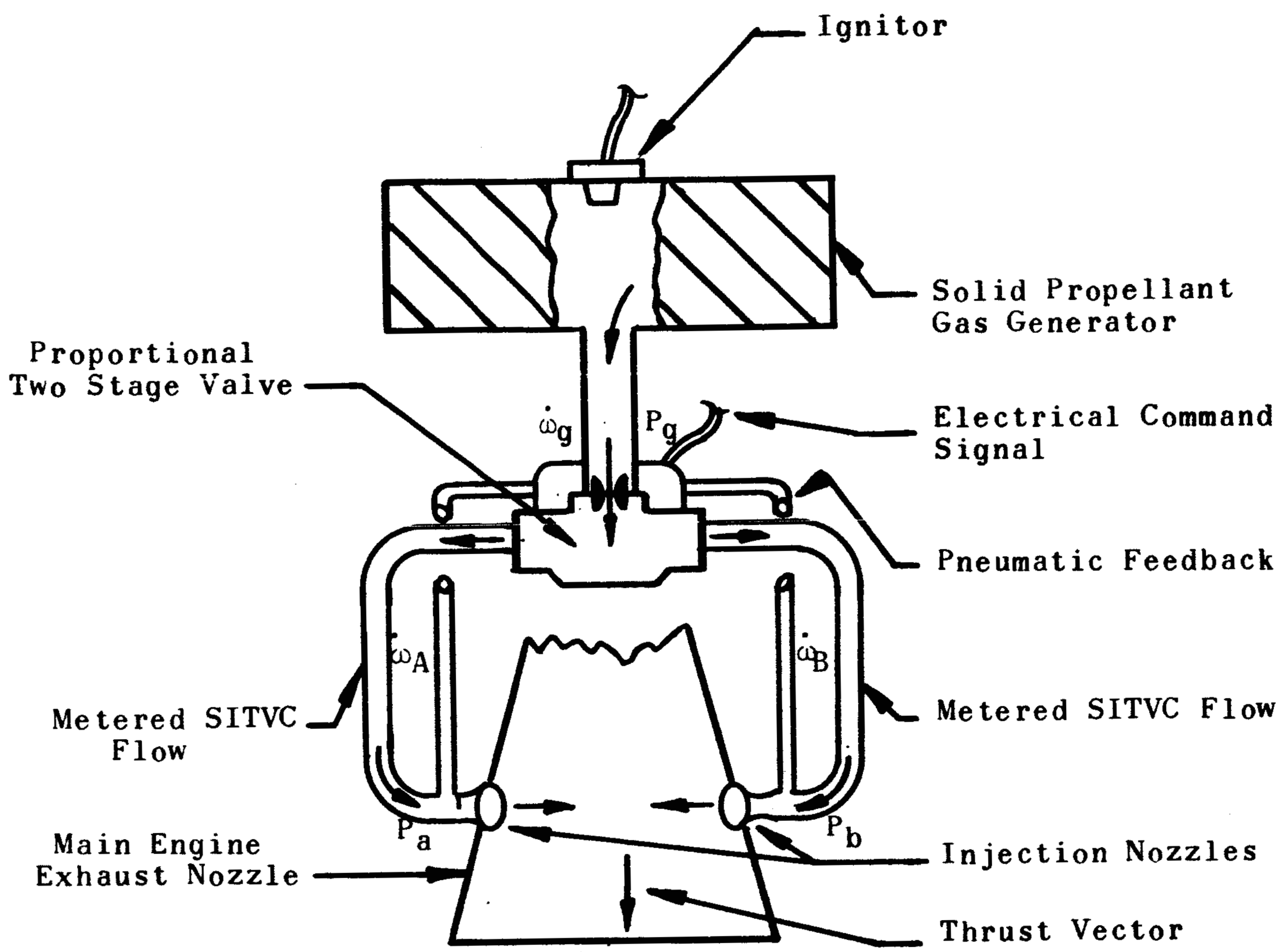
A schematic of the functional SITVC system is shown in Figure 2. Although the valve is applicable for vehicle altitude (pitch, yaw, roll) control, the system selected for this program was to demonstrate a single-axis secondary injector thrust vector control.

The SITVC single axis system is basically comprised of the solid propellant gas generator, two-stage pneumatic pressure feedback valve, output discharge manifolds and injection nozzles. The 2000 psig supply flow ($\dot{\omega}_g$) from the generator is maintained constant by use of a sonic-flow orifice located in the inlet of the two-stage valve. An electrical command signal to the torque motor controls the two-stage valve which provides proportional modulation of the output mass flow to the injection nozzles



VICKERS TWO-STAGE PRESSURE FEEDBACK VALVE

FIGURE - I

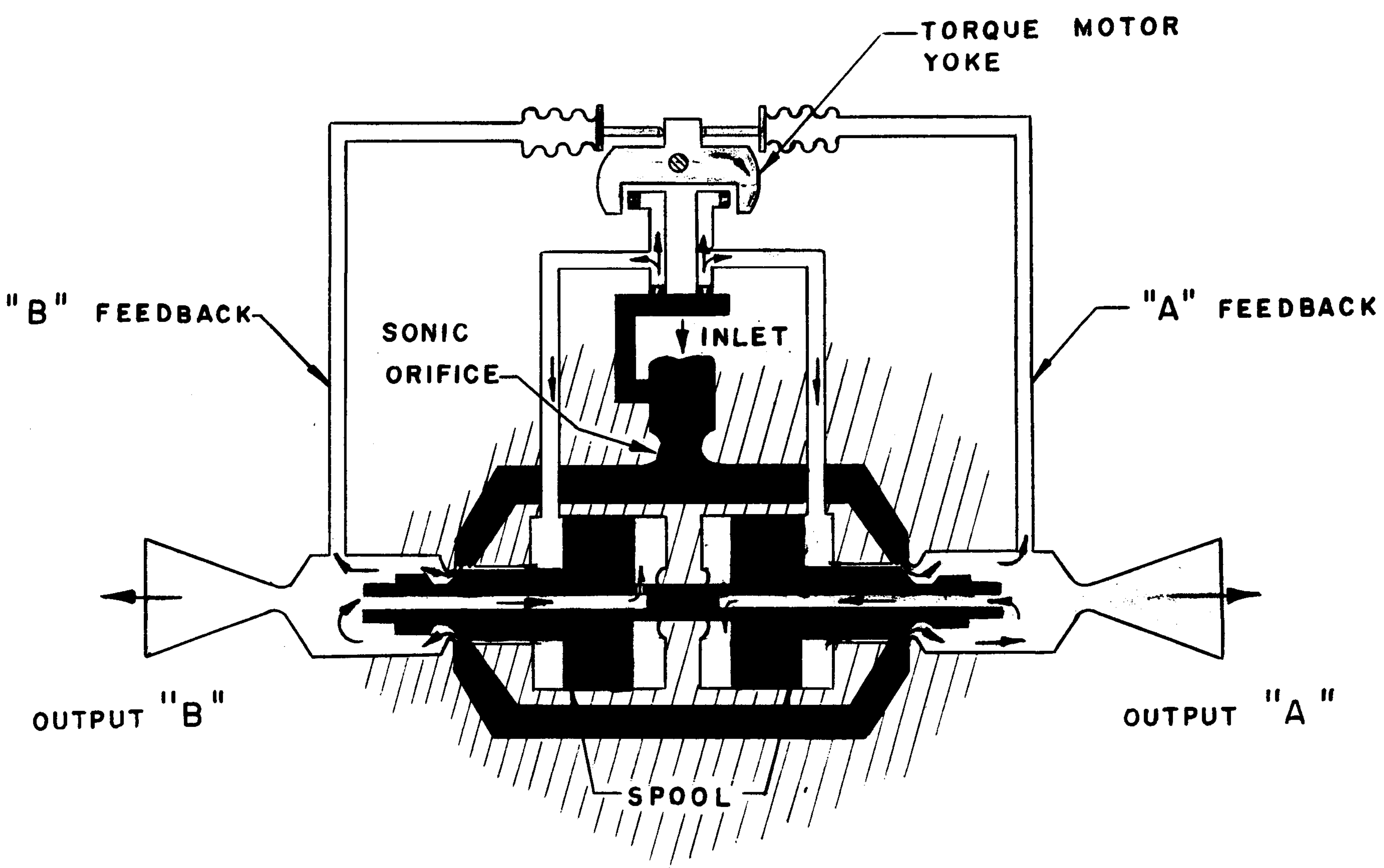


SINGLE AXIS SITVC SYSTEM

FIGURE 2

located in the main engine exhaust nozzle. The system loop is closed by pneumatic feedback of the differential injection pressure.

The two-stage valve is shown in schematic form in Figure 3. The staged valve consists of the main stage housing and metering spool, discharge pressure feedback bellows mechanism, a yoke type open center servo to control the servo stage orifices, and a proportional differential current multi-gap design torque motor. The valve utilizes the Type "O" servo control principle.



TWO STAGE PNEUMATIC SERVO VALVE SCHEMATIC

FIGURE 3

The electrical error command signal supplied, for example, by the vehicle's guidance system is transmitted to the electrical torque motor which produces a torque on the control beam proportional to the error signal. The control beam positions the yoke type flapper valve as a function of input torque minus the feedback torque (output chamber pressure differential times the feedback bellows area and lever arm). Positioning of the servo yoke causes a flow restriction on the servo orifice which creates a servo control pressure differential, which is applied to the net piston area of the main stage metering spool. The force created by the differential control pressures moves the main stage spool to the position providing the requested chamber pressure differential.

As shown in Figure 3, the output pressures are fed back to the input beam and translated in terms of negative torque feedback. This torque signal is subtracted from the input torque signal, thus closing the servo loop and resulting in the input error signal to the pneumatic servo.

To achieve a proportional output, pneumatic rate was incorporated in the main stage. This is accomplished by sensing the output pressure differential via a sensing tip probe on each end of the main spool and applying the output pressure differential to an effective spool area. The resulting differential force, which is linear with mass flow or spool position, results in a force balance condition on the main spool, thus creating a positive pneumatic spring rate.

SECTION 3

FLIGHTWEIGHT PROPORTIONAL TWO-STAGE VALVE DESIGN

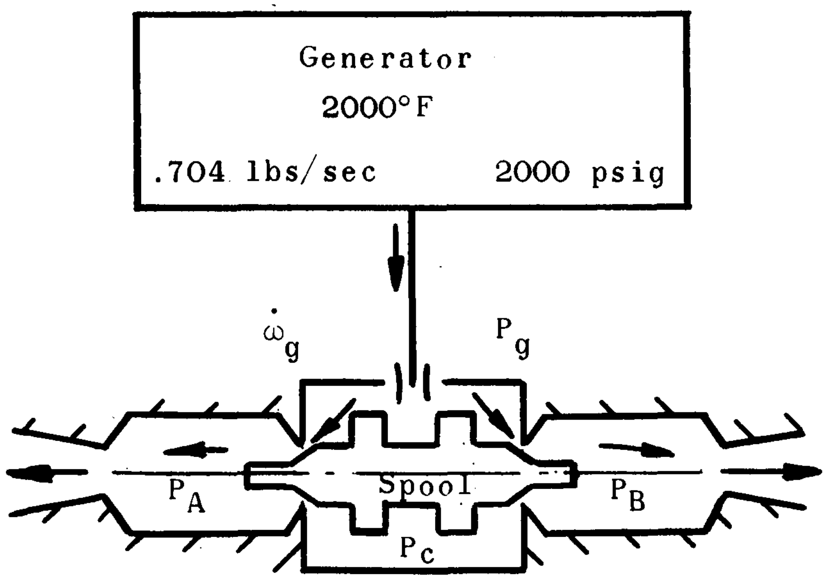
3.1 Analog Study and Design Parameters

An analog computer study was conducted using an EAI Pace 221 R and 231 R computer to optimize the design parameters and valve sizes to achieve the desired dynamic response performance and flightweight design criteria. This analysis is summarized in detail in Appendix B. The parameters which were critical to the design criteria were, torque motor spring rate, negative feedback area torque constant, main stage pneumatic spring rate, servo control volume, and the output manifold discharge volume. The pertinent system design requirements, parameters and sizes are shown in Figure 4.

Based on the analog study, valve stability was dependent on the torque motor spring rate, the negative feedback area-torque constant, main stage pneumatic rate and servo control volume. In addition, main stage pneumatic rate, servo control volume, and output manifold discharge volume were major parameters which affected dynamic response. The output manifold discharge volume appeared to be the main parameter limiting dynamic response.

Parameter-Size	Units	Design	Actual Hardware
P_g	psig	2000	2000
P_h	psig	1000	1000
P_A and P_B	psig	0-600	0-550
$\dot{\omega}_g$	lb/sec	0.704	0.704
Electrical Power	watts	6	5.5
Valve Weight	lbs	4.1	*4.24
Control Mode	---	Proportional	Proportional
Servo Capacity	%	3	3.1
Pneumatic Rate	lb/in	3600	2900
Feedback Torque Constant	in^3	0.0036	0.0035
Torque Motor Output	lb/in	3.6	3.0
Envelope	inches	Flight	2.5" sphere plus 3" x 1½" x 3" Torque Motor-Servo

*Includes Weight of Experimental Pressure Probes.



SYSTEM-DESIGN-REQUIREMENTS-PARAMETER-SIZES

FIGURE 4

Based on the finalized parameters and selected design criteria, (summarized below) the analog performance is shown in Figure 5. The theoretical design gains and dynamic response values for the two-stage valve are shown in Figure 6.

ANALOG STUDY SUMMARY

Design Criteria:	Inlet Pressure	2000 psig
	Inlet Temperature	2000°F
	Servo Capacity	3.0% of Total Flow
	Servo Leakage (assumed)	7.0% of Servo Flow
	Total Mass Flow	0.704 lb/sec
	Spool Friction	50 lbs
	Spool Stiction	60 lbs
	Pneumatic Rate	2900 lb/in
	Discharge Volume	20 inch^3 (total)
	Control Mode	Closed Loop

A function block diagram of the two-stage proportional pneumatic pressure feedback servo valve is shown in Figure 7. This diagram illustrates a graphical representation of the transfer function areas which yield the mathematical description of the proportional pneumatic valve and system dynamics. Non-linear equipment was utilized in the computer analysis to fully describe the inter-relationships of the defined parameters.

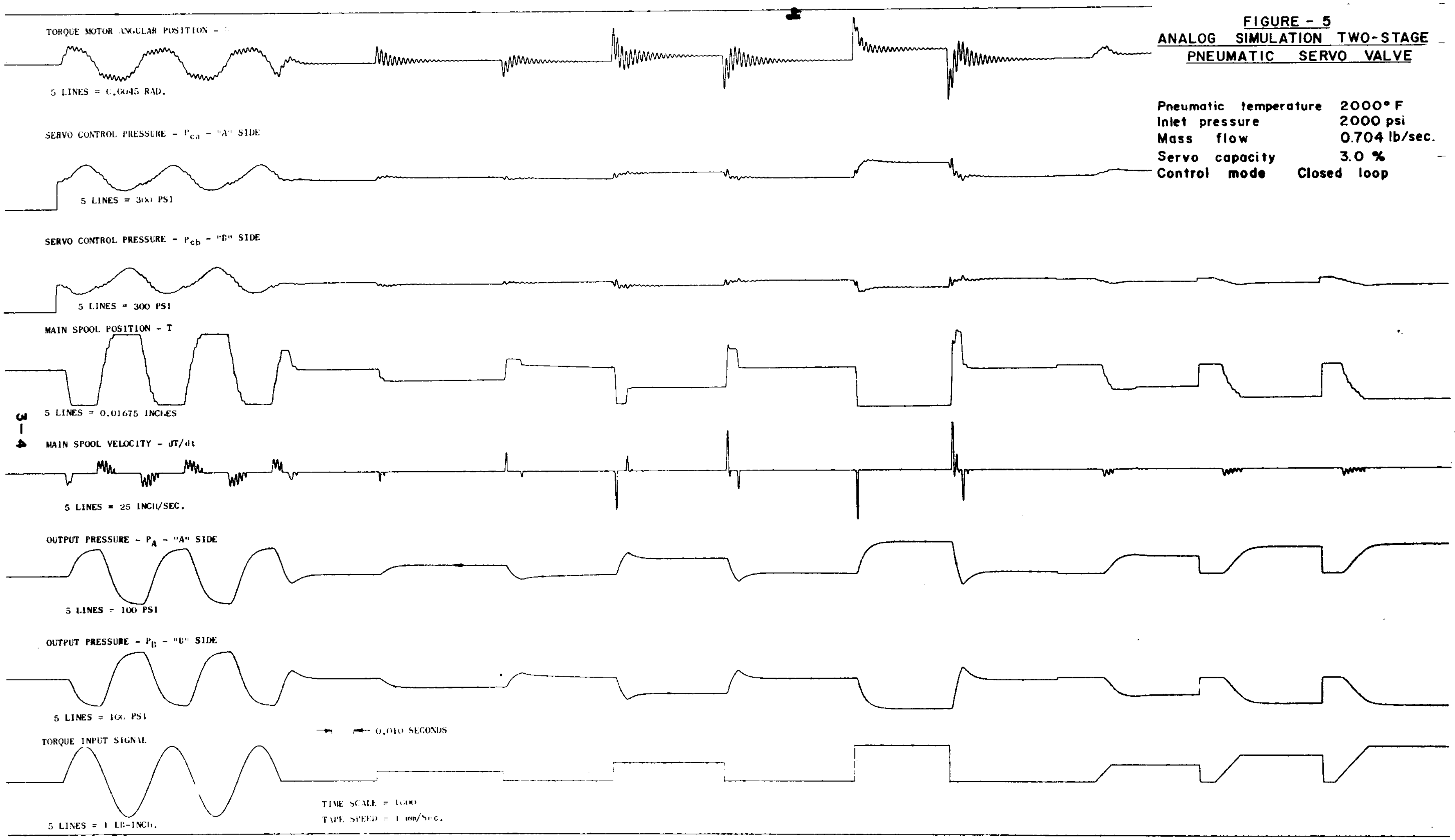


FIGURE - 5
ANALOG SIMULATION TWO-STAGE PNEUMATIC SERVO VALVE

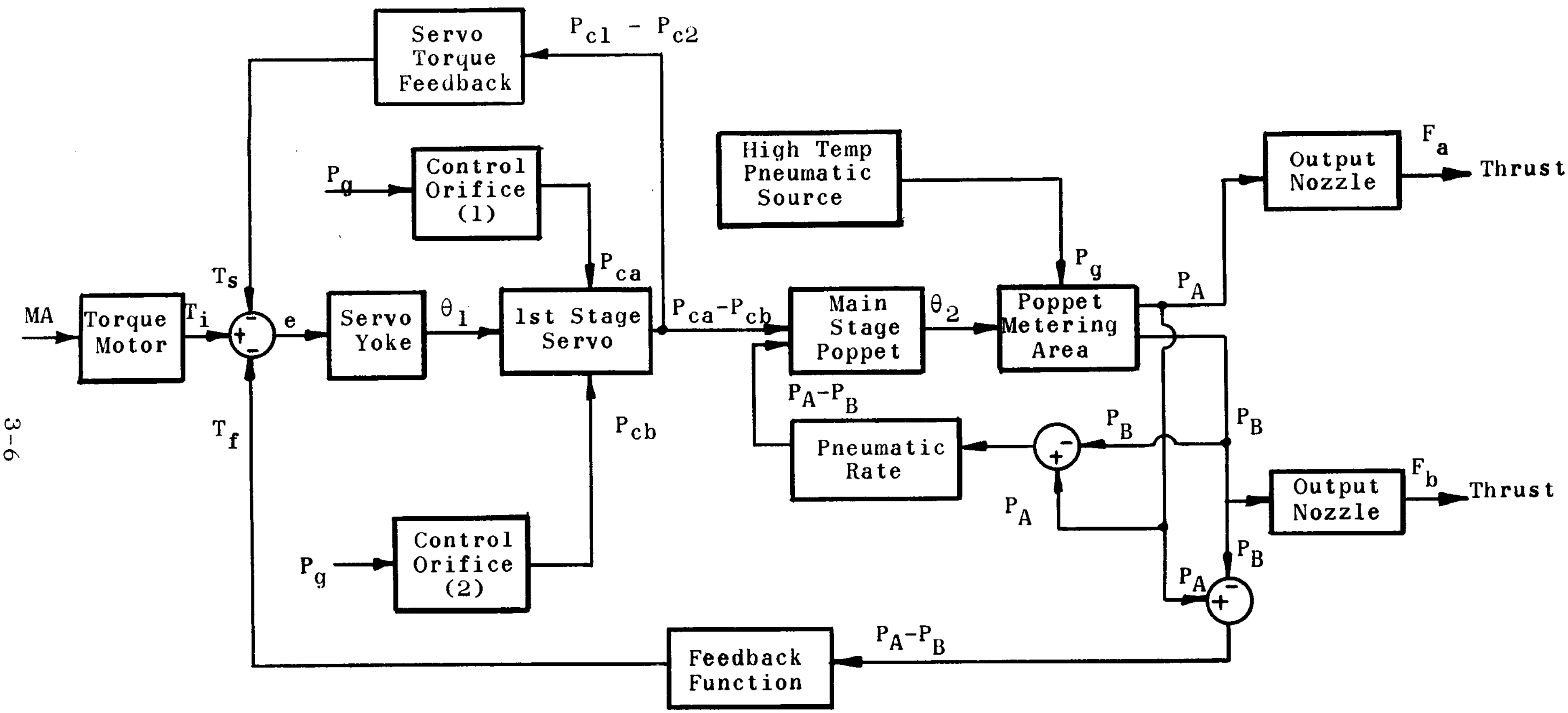
Pneumatic temperature	2000° F
Inlet pressure	2000 psi
Mass flow	0.704 lb/sec.
Servo capacity	3.0 %
Control mode	Closed loop

Parameters/Gain Response	Description		Value	Units
$\frac{(P_A - P_B)}{\Delta I}$	$\frac{\Delta \text{ Output Press}}{\Delta \text{ Input Current}}$	=	14.0	$\frac{PSI}{MA}$
$\frac{(P_A - P_B)}{(P_{cA} - P_{cB})}$	$\frac{\Delta \text{ Output Press}}{\Delta \text{ Servo Press}}$	=	1.2	$\frac{PSI}{PSI}$
$\frac{(P_{cA} - P_{cB})}{\Delta I}$	$\frac{\Delta \text{ Servo Press}}{\Delta \text{ Input Current}}$	=	12.0	$\frac{PSI}{MA}$
25% Command	Step Response			
(From Null)	Total Time		.009	sec
	Delay Time		.0015	sec
	Rise Time		.0075	sec
	% Overshoot		nil	%
	Settling Time (90% Final Value)		Zero	sec
50% Command	Step Response			
(From Null)	Total Time		.005	sec
	Delay Time		.0015	sec
	Rise Time		.0035	sec
	% Overshoot		33	%
	Settling Time (90% Final Value)		.007	sec
100% Command	Step Response			
(From Null)	Total Time		.0105	sec
	Delay Time		.0015	sec
	Rise Time		.009	sec
	% Overshoot		*	

*100% Command - valve spool hit "stops" on orifice seat preventing overshoot.

ANALOG DATA - TWO-STAGE VALVE

FIGURE 6



FUNCTION BLOCK DIAGRAM

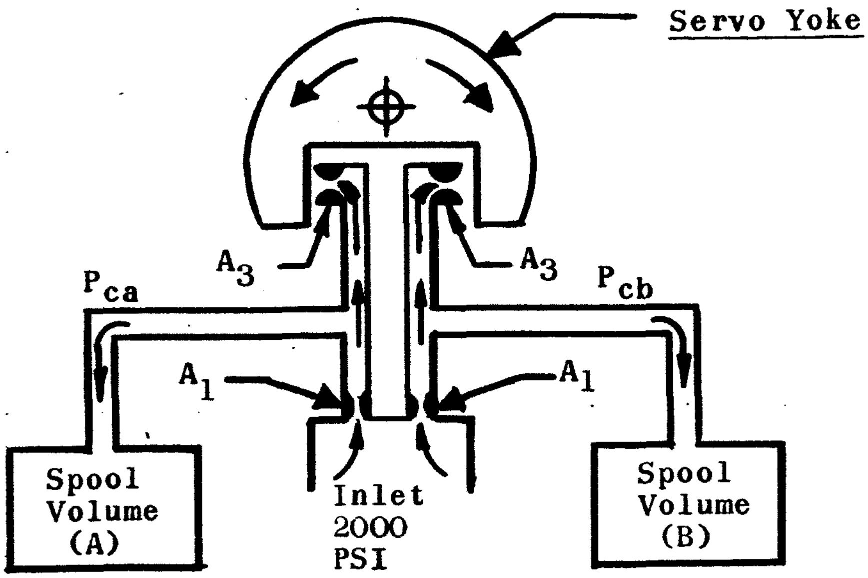
TWO-STAGE PROPORTIONAL HIGH TEMPERATURE PNEUMATIC VALVE

FIGURE 7

3.2 Servo Stage Design

Based on the design criteria of the analog study, a 3.0% servo design capacity was selected to realize the desired dynamic response performance. The servo driving function was a Vickers design linear differential current multigap torque motor. The torque motor mode of control was a push-pull circuit design. The torque motor was designed to provide a maximum output torque of 3.6 lb/in at full stroke (± .018 radians) using a 200 lb/in/RAD torque motor spring. A low moment of inertia (2.1 x 10^{-5} in/lb/sec^2) motor shaft and servo yoke assembly was designed which exhibited excellent damping within 12 cycles. The total electrical power consumption was 5.5 watts using a circuit sum current of 50 milliamps.

The 3.0% servo capacity utilized a thermal compensated yoke flapper design as shown in Figure 8.

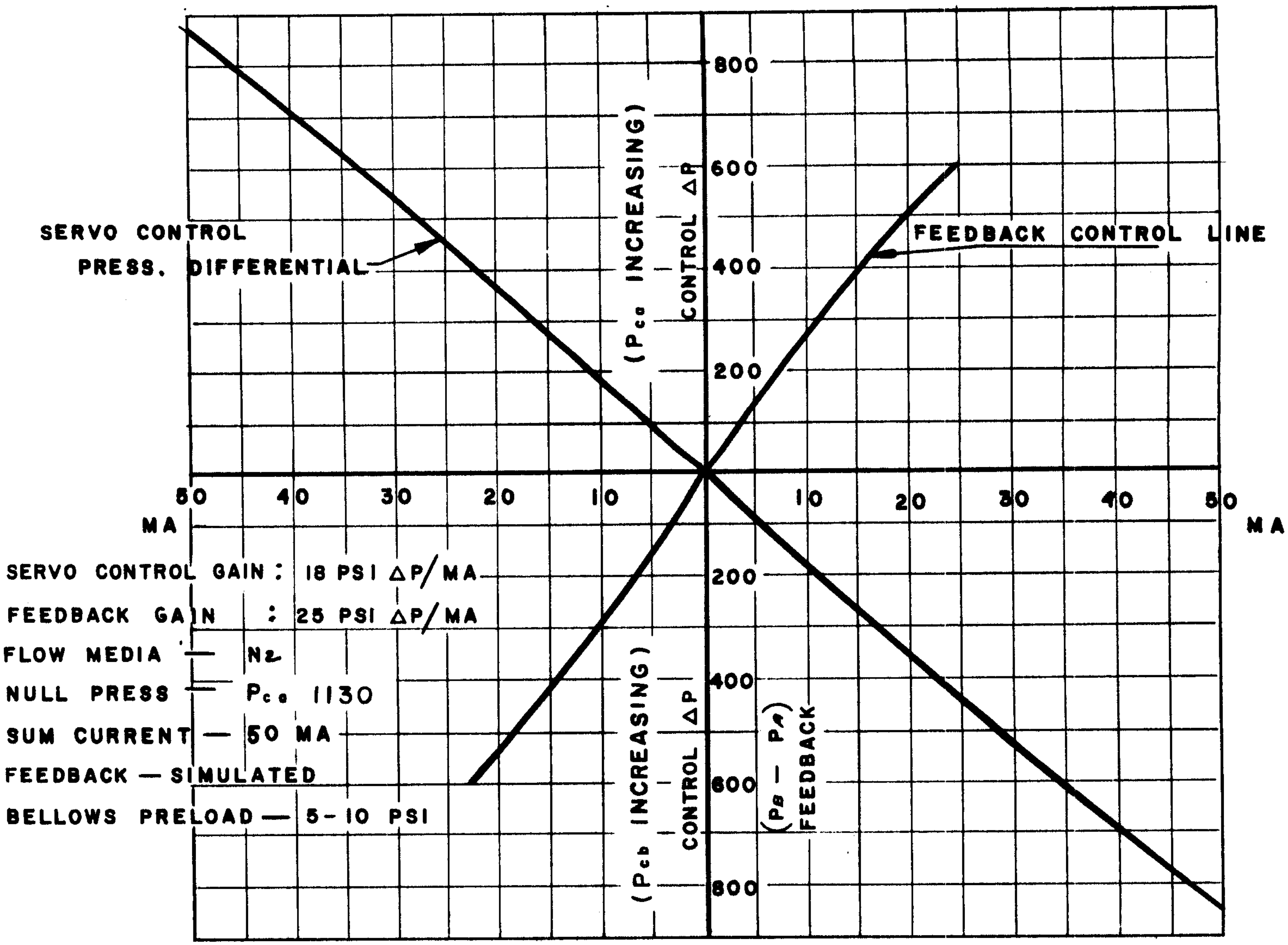


FIRST STAGE SERVO SCHEMATIC

FIGURE 8

A total mass flow of 0.022 lb/sec was utilized for servo flow to control 0.682 lb/sec main flow thus providing an amplification ratio of 31:1. Using an inlet pressure of 2000 psi, the up-stream servo orifices were sized to provide sonic or near sonic flow in the valve null positions. Based on the critical pressure ratio of .547 the null control pressure level (P_{ca} and P_{cb}) design would be 1100 psi. To achieve linear servo differential performance an $\alpha = \left(\frac{\text{area of metering orifice}}{\text{area of up-stream orifice}}\right)$ design ratio of 1.5 was used. The downstream fixed orifice area (A_3) in respect to the servo-yoke metering orifice area ratio was based on a design ratio of 3.0 minimum to prevent the A_3 area influence on the metering area. To minimize propellant exhaust contamination of the servo section, the smallest orifice used was .031 diameter (upstream orifice). It should be noted that 7% of the servo flow in the P_{ca} and P_{cb} passages was assumed lost due to leakage in the spool volume passages during cycling.

Movement of the servo yoke (Figure 8) will increase the servo pressure on (P_{ca} or P_{cb}) one side and will decrease the servo pressure on the adjacent side. Examination of the servo circuit readily shows sonic flow on one side and sub-sonic flow on the adjacent side of the servo in an "off-null" condition. This function produced a minimumized system non-linearity throughout the differential range as evidenced in Figure 9. In addition, the simulated feedback control characteristic is illustrated in Figure 9.



FIRST STAGE SERVO & SIMULATED FEEDBACK CONTROL CHARACTERISTICS (3% SERVO DESIGN)

FIGURE- 9

3.3 Output Stage Design and Pneumatic Rate

The output stage consisted of three basic housings and the double area metering spool. As the proportional servo pressure differential was applied to the spool servo area (reference Figure 3), proportional movement of the output stage spool was achieved based on the pneumatic rate. Pneumatic rate was achieved by sensing and applying the discharge pressure differential to the second spool effective area in such a manner that the resulting proportional force would oppose the servo force thus creating a proportional force balance position for each selected input signal magnitude.

The method of achieving a constant pneumatic rate was entirely dependent on the spool tip design since high velocity pneumatic flow could produce an aspiration effect on the pneumatic rate feedback pressure. This problem was studied in detail and is presented in Section 4.1.

A sharp edge metering orifice design was used in the main stage spool to orifice metering area. Metering was accomplished in the sonic flow regime.

3.4 Closed Loop Pneumatic Feedback

To achieve static accuracy and system dyanmic performance, the system loop was closed by utilizing pneumatic feedback. As shown in Figure 3, the discharge pressure differential is fed back and applied to a dual bellows mechanism. The pressure differential is converted to force times a lever arm to provide negative feedback torque to the torque motor spring yoke assembly. The functional application of the feedback bellows mechanism was previously demonstrated utilizing 5600°F pneumatics at Vickers Incorporated. The application of bellows utilizing the low thermal conductivity stagnated gas barrier principle was applied to the feedback mechanism design. Thus, bellows operating temperature should not exceed 600°F. Since the bellows spring rate was a negligible factor in the design equations, a loss of bellows rate due to temperature effect would produce a negligible change in the operating characteristic. In addition, closed loop design provided a capability of accurate control and dynamic response independent of extraneous internal valve forces and output stage valve friction by utilizing the relatively large control forces available from first stage force amplification. Since servo force amplification was utilized in a closed loop mode of operation, main stage flow force effect would be minimized and should only contribute a non-linearity at the extreme end positions of the spool travel.

3.5 2000°F Dynamic and Static Seals

The piston ring dynamic seal design which was previously proven under contracts NAS 1-2962 and NAS 8-5041 was selected for the two-stage valve design. High temperature alloy ductile iron-parco lubrized outer rings with a inconel "X" inner expander piston ring were used with a design radial clearance of .0025 inches. This combination seal was incorporated in all valve dynamic seal areas with the exception of the spool center shaft seal. The center shaft seal was a single ductile iron-parco lubrized ring. Both the single and two piece ring seal assemblies were purchased from Koppers Company, Metal Products Division, Baltimore 3, Maryland. All dynamic seals were used in Stackpole 331 graphite bores.

Gold plated inconel "X" metal static "Vee" design seals were selected for all valve critical static seals. Raybestos Manhattan A-56 gasket material was used at the interface of the valve outlet to discharge manifold. The metallic "Vee" seals were purchased from Parker Seal Company, Culver City, California.

3.6 Materials of Construction

The three main stage housings of the two-stage valve were fabricated from Latrobe BG-42 high temperature martensitic steel. RA-330, a super alloy austenitic grade stainless steel was used for the main stage spool, metering orifice seats, discharge nozzles and all flow orifices (except the servo downstream orifices). Selection of this material was based on excellent erosion resistance performance under 2000°F conditions in fixture tests conducted by Vickers Incorporated. Stackpole 331 graphite material was used for all dyanmic seal bores. Hastaloy "C" tubing was utilized for fabrication of the discharge manifolds. The servo stage housing was fabricated from 303 stainless steel since it was thermally insulated from the main housing using grain oriented pyrolytic graphite material for a heat shield. The torque motor shaft, servo yoke, and servo downstream orifice block were fabricated from a molybdenum (M_o-0.5T_i) alloy to utilize the low coefficient of thermal expansion characteristic. A welded construction bellows fabricated from Inconel "X" was used for the feedback mechanism. The bellows were purchased from Metal Bellows Corporation, Sharon, Massachusetts. The Vickers designed torque motor was constructed from standard equipment materials. The magnets were Alnico #5, armature and pole laminations were M-6X steel, Aluminum frame, and insulated magnet wire was used for the coils.

3.7 Solid Propellant Gas Generator

The solid propellant gas generator used for all high temperature test was an existing design used on previous NASA contracts. A fully developed grain formulation, "OMAX 453 D", made by Olin Mathieson was used to provide the 2000°F pneumatic source to the test appratus.

The generator consists of an ignitor, breech, two cured propellant grains, insulation, and a burst diaphram assembly.

The breech contains the two solid propellant ammonium nitrate end-burning grains which burn concurrently, producing a gas mass flow of 0.704 lbs/sec at 2000 psig pressure with a flame temperature of 2000°F. Pertinent ballistic properties and information is tabulated below:

Symbol	Description	Value	Units
R	Gas Constant	80.3	ft/°R
K	Ratio of Specific Heat	1.279	--
M	Gas Molecular Weight	19.25	lbs.
t	Total Burn Duration	43 (NOM)	Seconds

SECTION 4

SYSTEM TEST AND FLIGHTWEIGHT TWO-STAGE VALVE PERFORMANCE

4.1 Pneumatic Rate - Schlieren Study

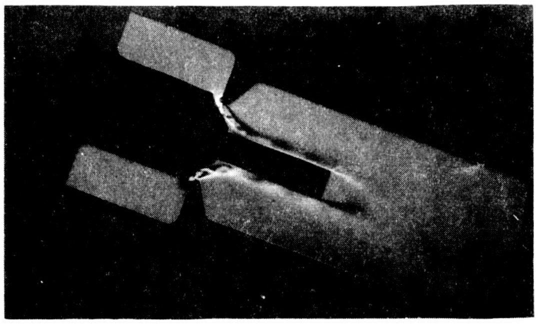
Early in the program, the main spool tip design was studied using a Schlieren two-dimensional scale model to determine the best tip configuration for achievement of pneumatic rate. The design criteria was to sense the equivalent sub-sonic flow discharge static pressure for application to the applicable spool areas. Analysis of the flow profile along the spool axis showed that the pneumatic velocity would be sonic and probably supersonic at the area where the pressure feedback tap was desired. Thus, a spool tip configuration was developed to provide a pressure probe downstream of the high pneumatic velocity metering orifice section. Figure 10 shows the Schlieren flow jet profile designs tested. Several projection designs were tested to achieve a constant shock projection independent of the spool position. As expected, a 3/4 inch long probe would aspirate providing a somewhat non-linear rate ranging from 40 to 72% of the discharge static pressure dependent on spool position.

The best design selected, based on the two dimensional Schlieren study, was a .030 height projection by .040 wide located at the center of the 3/4 inch probe. This design created a pneumatic shock wave, decreased flow to the sub-sonic regime, and provided approximately 93% of the discharge static pressure for pneumatic

PNEUMATIC FLOW JET PROFILE DESIGNS

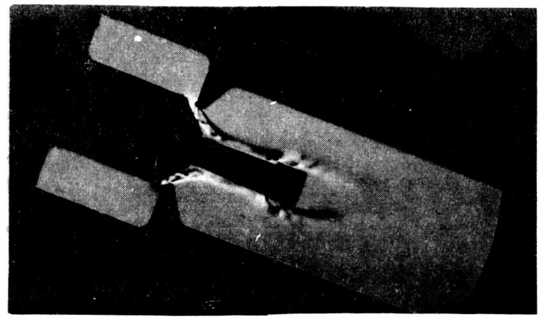
SCHLIEREN TWO DIMENSIONAL FULL SCALE MODEL

PRESSURE RATIO — 4:1 NULL STROKE —.066



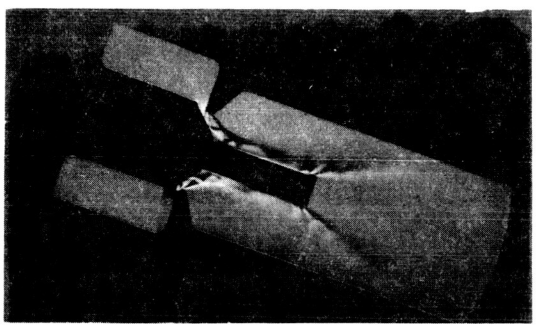
PLAIN TUBE

NO SHOCK PROJECTION



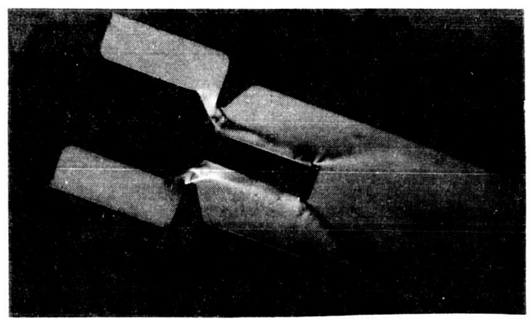
SHOCK PROJECTION

END— .025 x .25 LONG



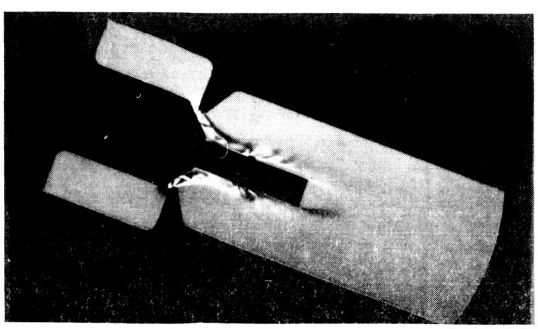
SHOCK PROJECTION

END—.015 x .040 LONG



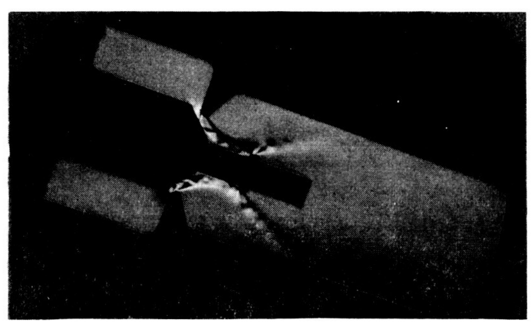
SHOCK PROJECTION

END—.025 x .040 LONG



SHOCK PROJECTION

CENTER—.015 x .040 LONG



SHOCK PROJECTION

CENTER—.030 x .040 LONG

FIGURE 10

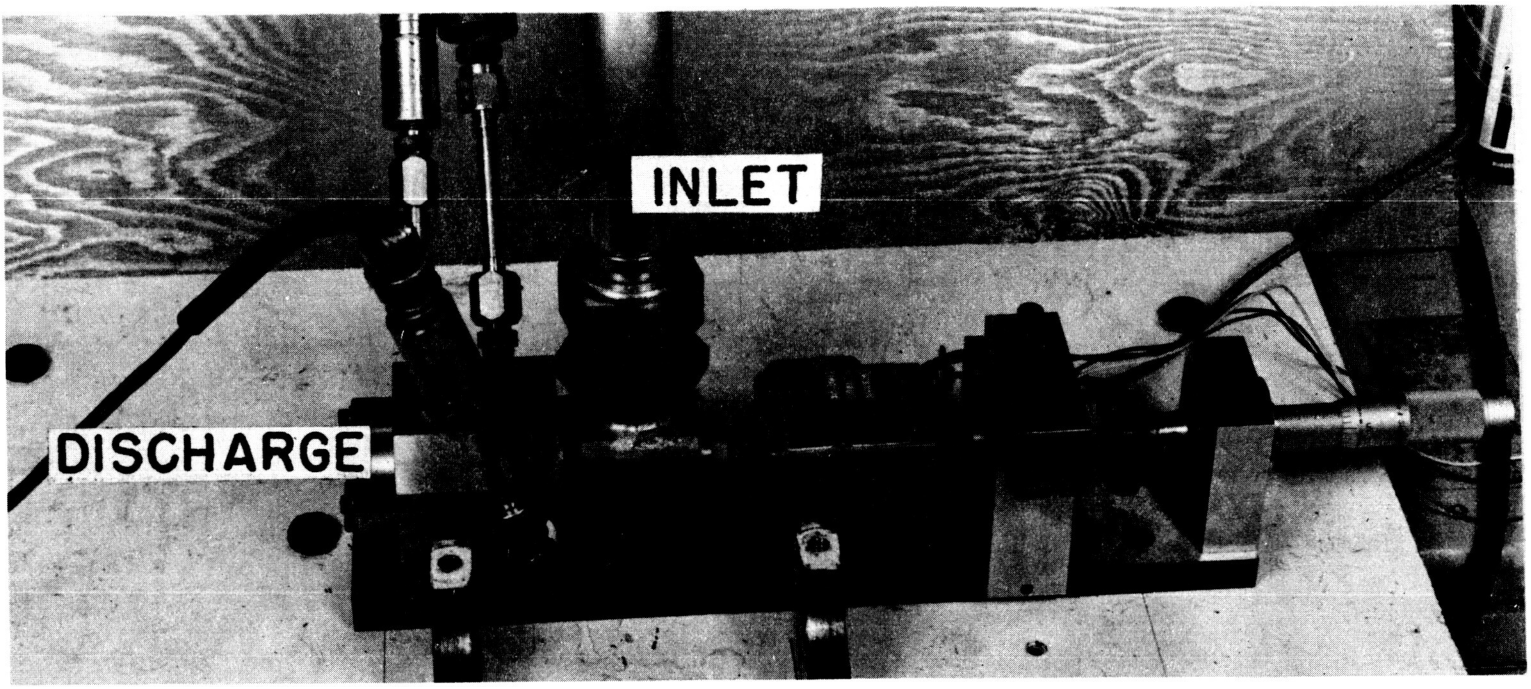
rate feedback.

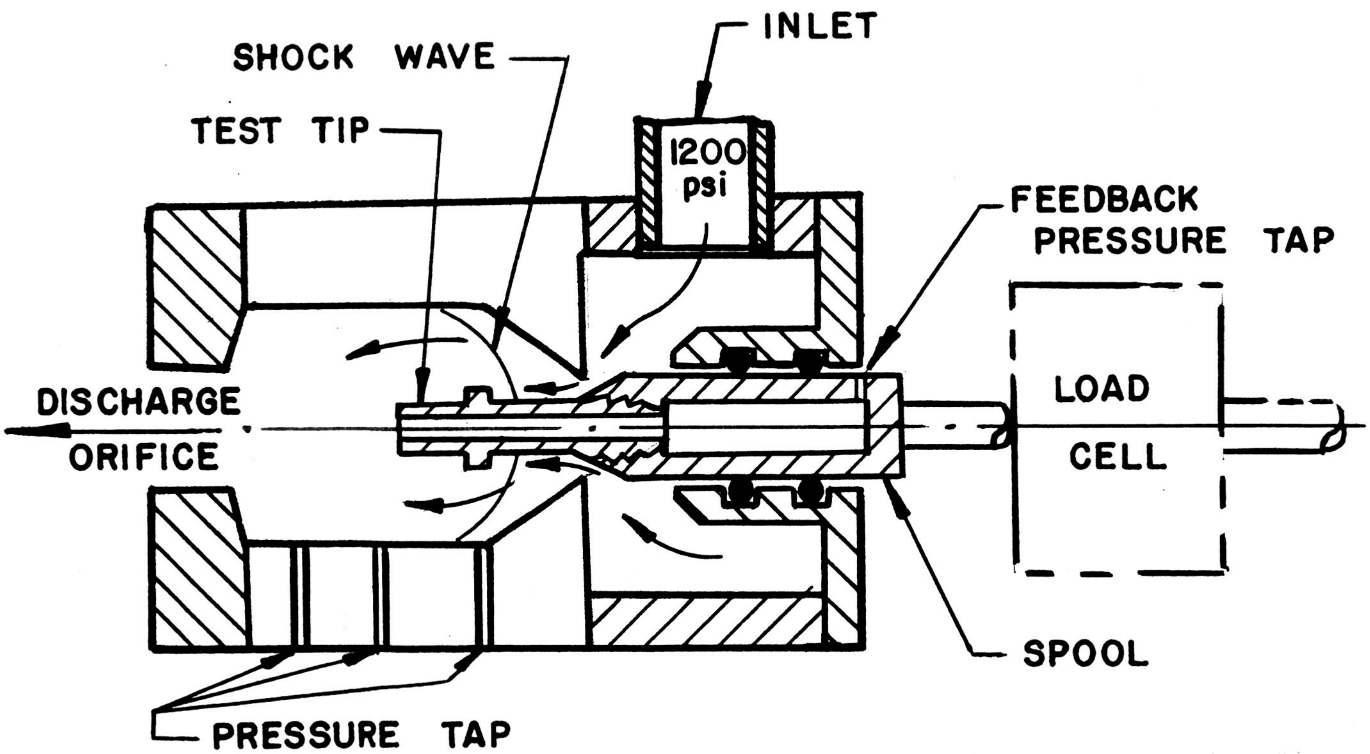
Since this valve development area was critical to desired valve performance, full scale three-dimensional fixture hardware was fabricated as shown in Figure 11.

This fixture was also equipped with a load cell which permitted the measurement of spool pneumatic flow forces. Four basic tip designs were tested and the results of these tests are shown in Figure 12. As previously stated, the design criteria to realize a constant pneumatic rate would be 100% pressure recovery. The design center shock projector selected from the Schlieren study was verified. Development of the contour tip design was developed later in the program and is discussed in Section 4.2.

4.2 Spool Stabilization, Flow Forces, and Achievement of Design Gains

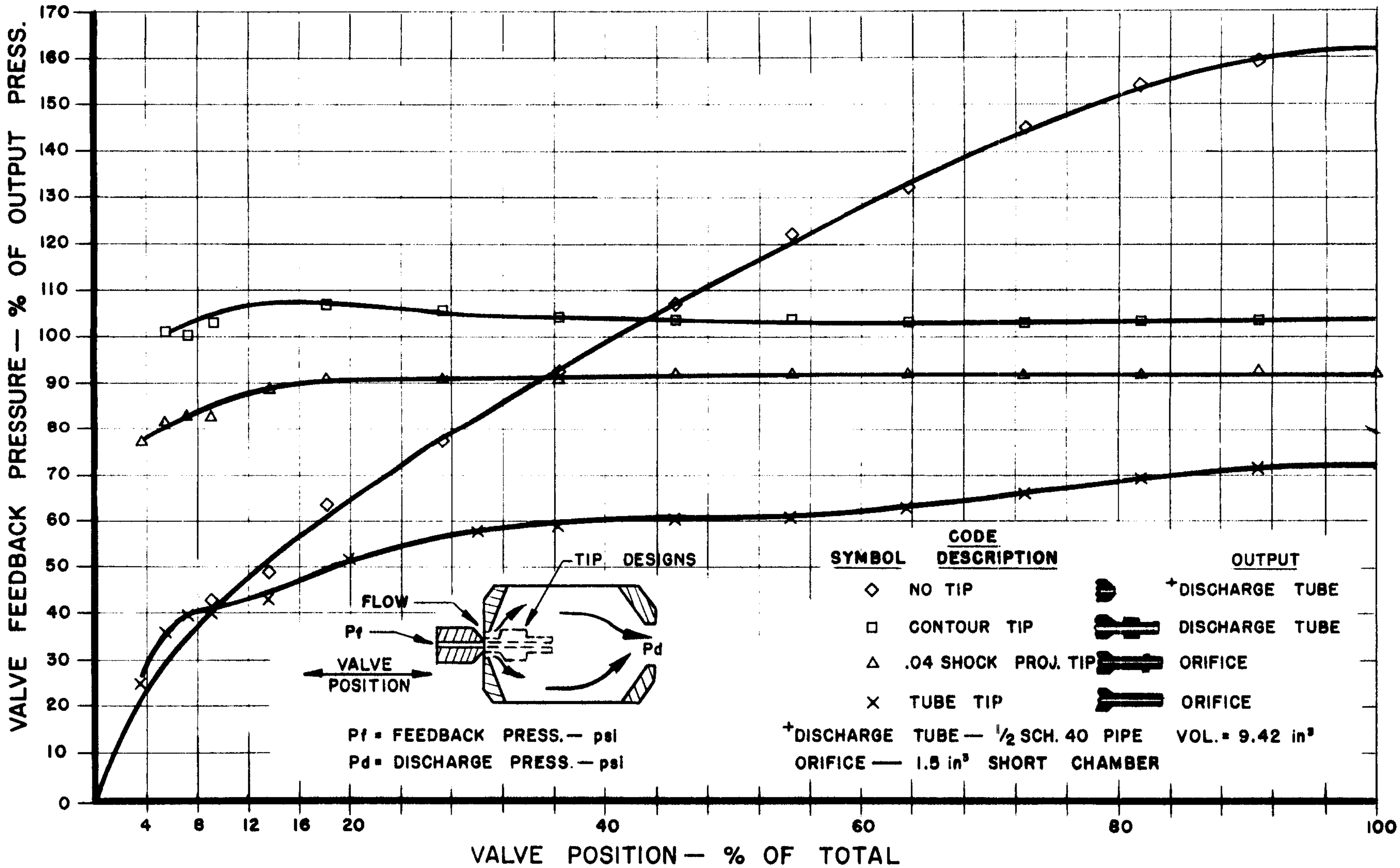
The center shock projection tip design when incorporated in the two-stage valve hardware exhibited marginal valve stability, sensitivity to step input signal amplitude and showed evidence of a limit cycle around the null position. Based on gaseous nitrogen testing, the experimental valve design gains were not in accordance with the theoretical design gains. It was summarized that the pneumatic shock wave from the spool tip was not consistent throughout the valve stroke. In addition, since pneumatic flow force is proportional to velocity, and examination of the Schlieren photographs indicated super-sonic





PNEUMATIC TEST FIXTURE

FIGURE - 11



EFFECT OF SPOOL TIP DESIGN ON PNEUMATIC FEEDBACK PRESSURE

FIGURE - 12

flows along the spool tip, the tip design was altered to provide a "contoured" tip design with the shock projection located as close as possible to the spool metering orifice without affecting the metering characteristics. This design would reduce the magnitude of the velocity component applicable to the flow force equation. The contoured design was tested in the flow force test fixture and exhibited a valve pneumatic rate feedback pressure of approximately 103% (see Figure 12). The 3% discrepancy is due to discharge manifold tube losses when comparing feedback pressure level to discharge pressure at the injection nozzle. It should be noted that when no tip configuration was tested that a valve feedback pressure of 162% of discharge pressure was observed. Due to the design and position of the spool tip, the "no tip" design provided a method of tapping the pressure gradient across the face of the metering spool, thus resulting in the relatively high percentage of valve feedback pressure.

The contour tip design was tested under gaseous nitrogen conditions and resulted in achievement of the valve design gains and excellent valve stability. A tabulation of the design gains versus tip design is shown in Figure 13.

Based on spool force measurements using the pneumatic test fixture, the total spool forces were determined for three selected pressure ratios. A mathematical analysis of the spool forces under identical fixture conditions permitted

A. Theoretical

Closed Loop	$\Delta P_d/\Delta P_c$	$\Delta P_d/\Delta ma$
	1.2	12

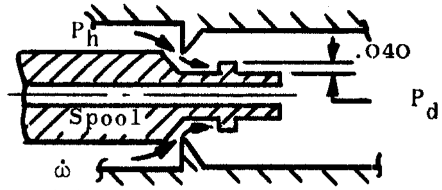
where: $\Delta P_d/\Delta P_c$ = Spool Pressure Gain = $\dfrac{\Delta \text{ Output Pressure - psi}}{\Delta \text{ Control Pressure - psi}}$

$\Delta P_d/\Delta ma$ = Valve Gain = $\dfrac{\Delta \text{ Output Pressure - psi}}{\Delta \text{ Input Current - ma}}$

B. Spool Tips With .040 Shock Projection

	$\Delta P_d/\Delta P_c$	$\Delta P_d/\Delta ma$
Closed Loop - Gaseous N_2	$\frac{470}{300} = 1.57$	$\frac{470}{40} = 11.75$
3rd Valve Unit - .128 Stroke	$\frac{610}{400} = 1.52$	$\frac{610}{54} = 11.3$
1/8" Size FB Lines	$\frac{240}{150} = 1.60$	$\frac{240}{19} = 12.6$

Equivalent Hot Mass Flow = 0.704 lb/sec.

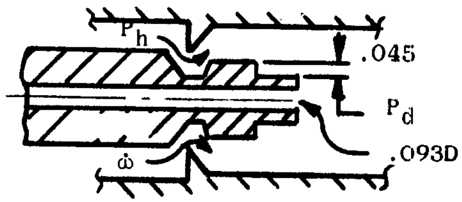


Stability - Marginal unstable to Step Inputs Limit Cycle at Null

C. Spool Tips With Contoured Shock Projection

	$\Delta P_d/\Delta P_c$	$\Delta P_d/\Delta ma$
Closed Loop - Gaseous N_2	$\frac{100}{100} = 1.0$	$\frac{100}{10} = 10$
3rd Valve Unit - .128 Stroke	$\frac{245}{225} = 1.09$	$\frac{245}{25} = 9.7$
1/8" Size FB Lines	$\frac{520}{420} = 1.24$	$\frac{520}{47} = 11.1$
	$\frac{560}{500} = 1.12$	$\frac{560}{59} = 9.5$

Equivalent Hot Mass Flow = 0.704 lb/sec.



Stability - Excellent

THEORETICAL AND EXPERIMENTAL VALVE GAIN VERSUS SPOOL TIP GEOMETRY

FIGURE 13

determination of the valve forces on a theoretical basis. Thus, a discrepancy between theoretical and experimental corrected data would define the magnitude of the pneumatic flow force. As shown in Figure 14, the experimental force data closely agrees with the theoretical force data. Thus, pneumatic flow forces appear to be minor (less than six pounds).

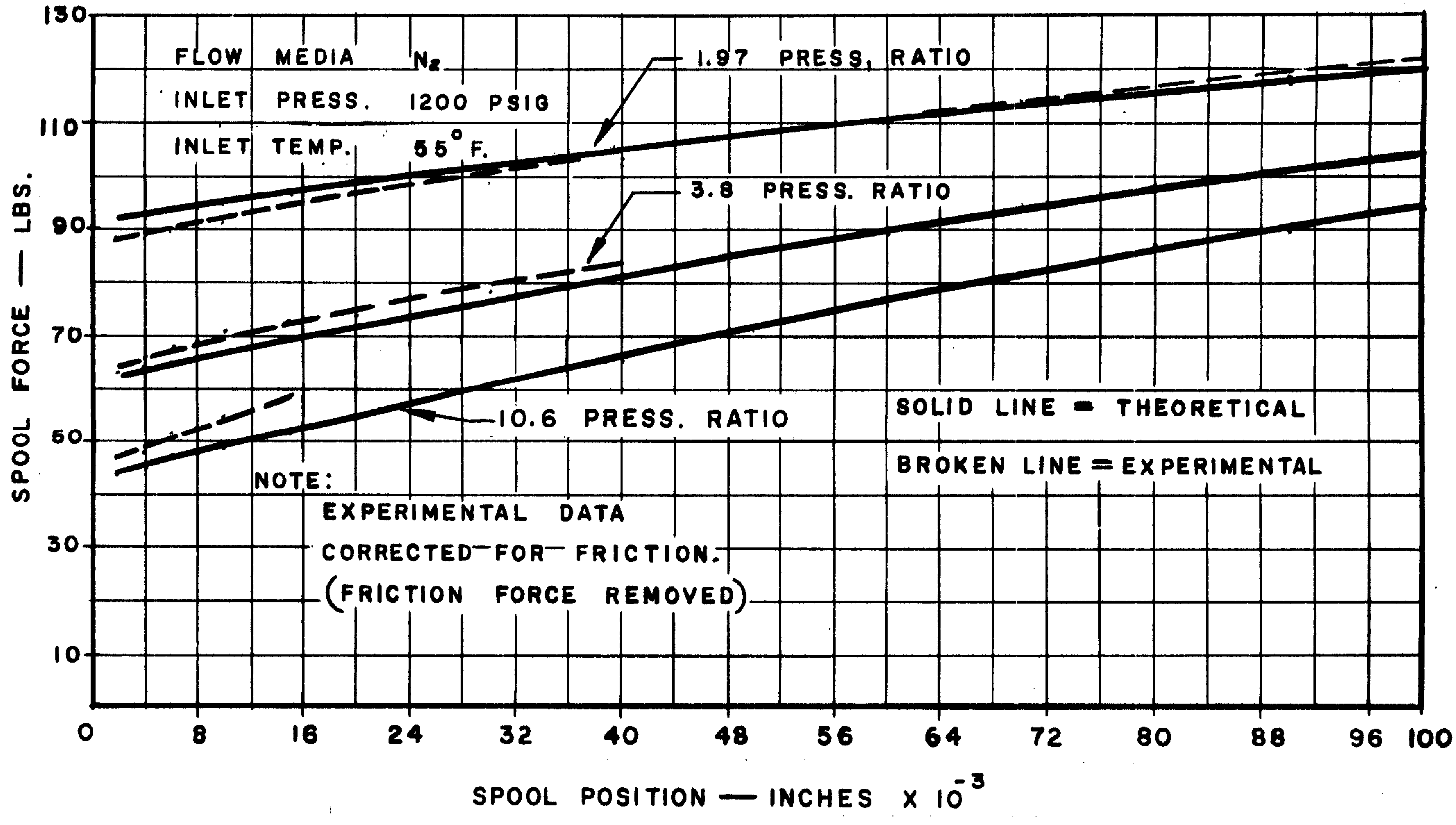
4.3 High Temperature Pneumatic (2000°F) Performance

Six (6) high temperature pneumatic test firings were conducted on the test stand, shown in Figure 15, at Vickers Incorporated, Troy, Michigan. All tests were conducted in a simulated SITVC altitude using 10 cubic inch volume discharge manifolds.

All system 2000°F tests were preceded by gaseous nitrogen tests for calibration and valve performance verification. Inlet, servo pressures, valve case pressure, and output discharge pressures were monitored in each test. In addition, input signal current was recorded. The gaseous nitrogen test data was used for correlation with 2000°F valve performance. All high temperature tests were conducted in a closed loop mode of control.

2000°F Test #1

The purpose of this test was to determine the system steady-state performance and system proportionality. The valve failed immediately following ignition which provided a minimum of meaningful test information. A fracture occurred across the main servo section inlet which prevented servo operation. The

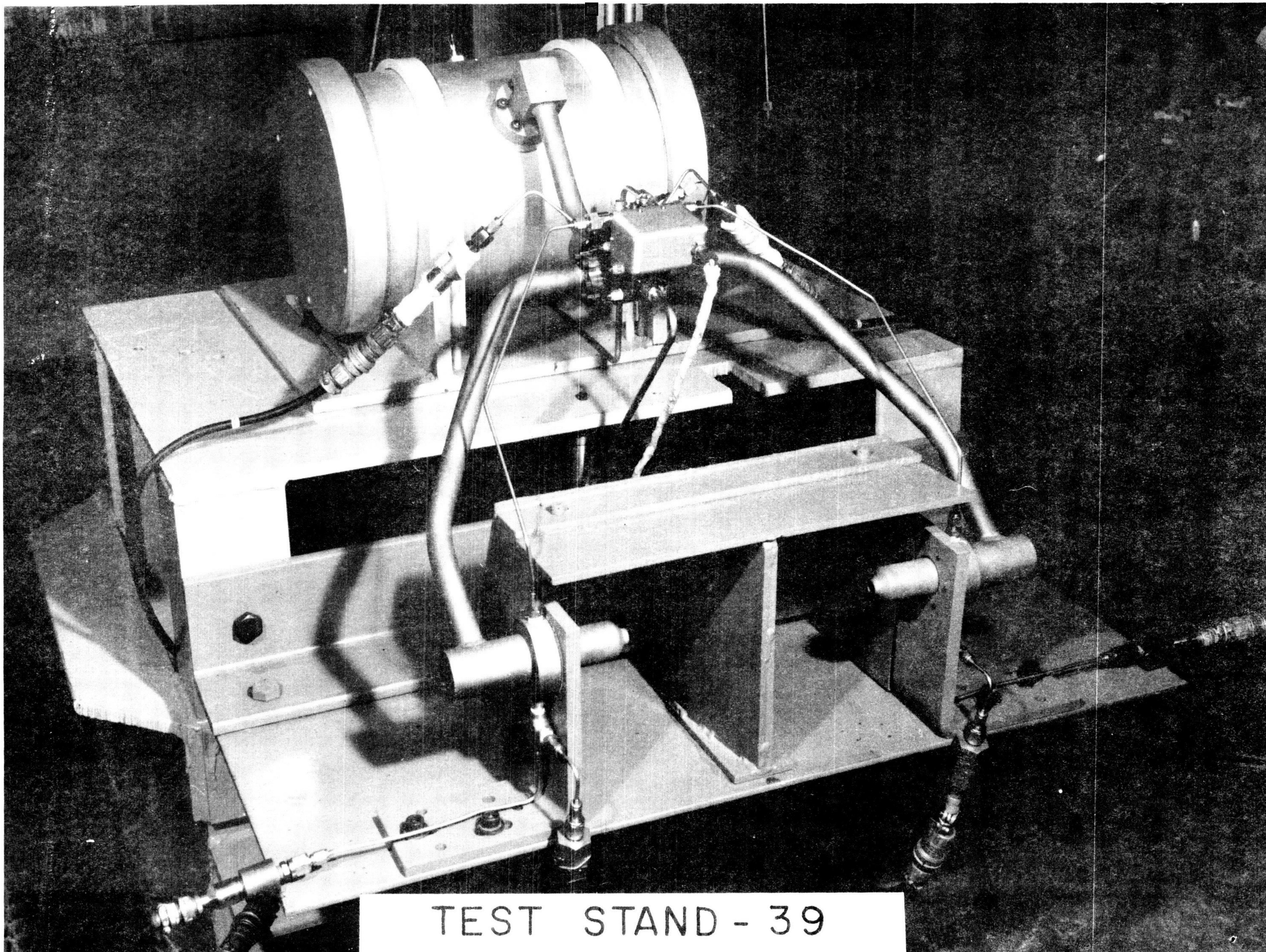


SPOOL FORCE VS. SPOOL POSITION

45° HALF ANGLE NOSE AND ORIFICE DESIGN

SINGLE SPOOL PNEUMATIC VALVE CONFIGURATION 100891-X

FIGURE - 14



TEST STAND - 39

FIGURE - 15

initial design of this particular unit was to use piston rings on a hardened steel bore. Although the test was a failure, two potential problem areas were projected:

(1) thermal sensitivity of the valve material at the desired hardness level for piston ring bore designs,

(2) visual evidence of dynamic seal failure potential with the selected ring-bore design.

The problem solutions projected were:

(1) annealing of the valve hardened parts to prevent thermal sensitivity, and

(2) incorporation of graphite bushings for the metal piston ring dynamic seals.

<u>2000°F Test #2</u>

This valve unit was reworked to incorporate the projected solutions from test firing #1. The objective of test #2 was to define steady state performance and system proportionality. This test demonstrated acceptable thermal sensitivity and dynamic sealing of the valve throughout the forty-three second test duration. Analysis of the test data and visual inspection of the valve hardware following the test showed a control instability problem upon ignition and a loss of servo control. This valve contained the center shock projection tip design which was proven to be marginally stable on valve #3. Inspection of the hardware showed that the servo section shaft which transmits servo position commands froze, thus preventing activation of a control pressure differential to the main stage spool. However, the thermal sensitivity and dynamic seal problem areas were corrected.

2000°F Test #3

Since test firing #2 indicated a control instability problem, analysis and extensive experimental investigations of the spool shock projection geometry dictated a design modification of the spool tip to achieve acceptable stability and valve design gain performance. The contoured tip design was incorporated in this unit. In addition, the servo yoke and shaft were modified to incorporate a shield to protect the shaft bearings and to increase the servo stroke.

Excellent high temperature test data was obtained for the first twenty-one seconds of the 43 second duration test. As can be noted in response traces, presented in this section of the report, low valve noise to input signal ratio characteristics were realized. In addition, valve sensitivity was exceptional in that valve output performance responded to imperfections in the "taped" input signal program schedule. At twenty-one seconds, one of the discharge manifolds separated from the two stage valve discharge flange and impacted the spool shock tip, thus preventing further control of the valve. As confirmed by this test, good control stability was achieved. The projected solutions to the manifold failure problem was to eliminate the fixed valve mounting and to mount the valve in a free suspension mode.

2000°F Test #4

An attempt was made to increase the servo control pressure level on the fourth unit to average the servo control decrease observed during previous tests. As a result, the servo froze which prevented valve control throughout the 44 second test. Visual examination of the hardware after test showed a failure of one of the graphite bushings. Failure of the graphite bushing was attributed to a pressure buckling of the metal support shell at the temperature operating stress level. The projected solution of the marginal structure problem was to re-design the valve end cap geometry to provide a stronger metal support shell for the graphite. The free suspension mounting of the valve proved successful, and no other structural problems were observed or experienced.

2000°F Test #5

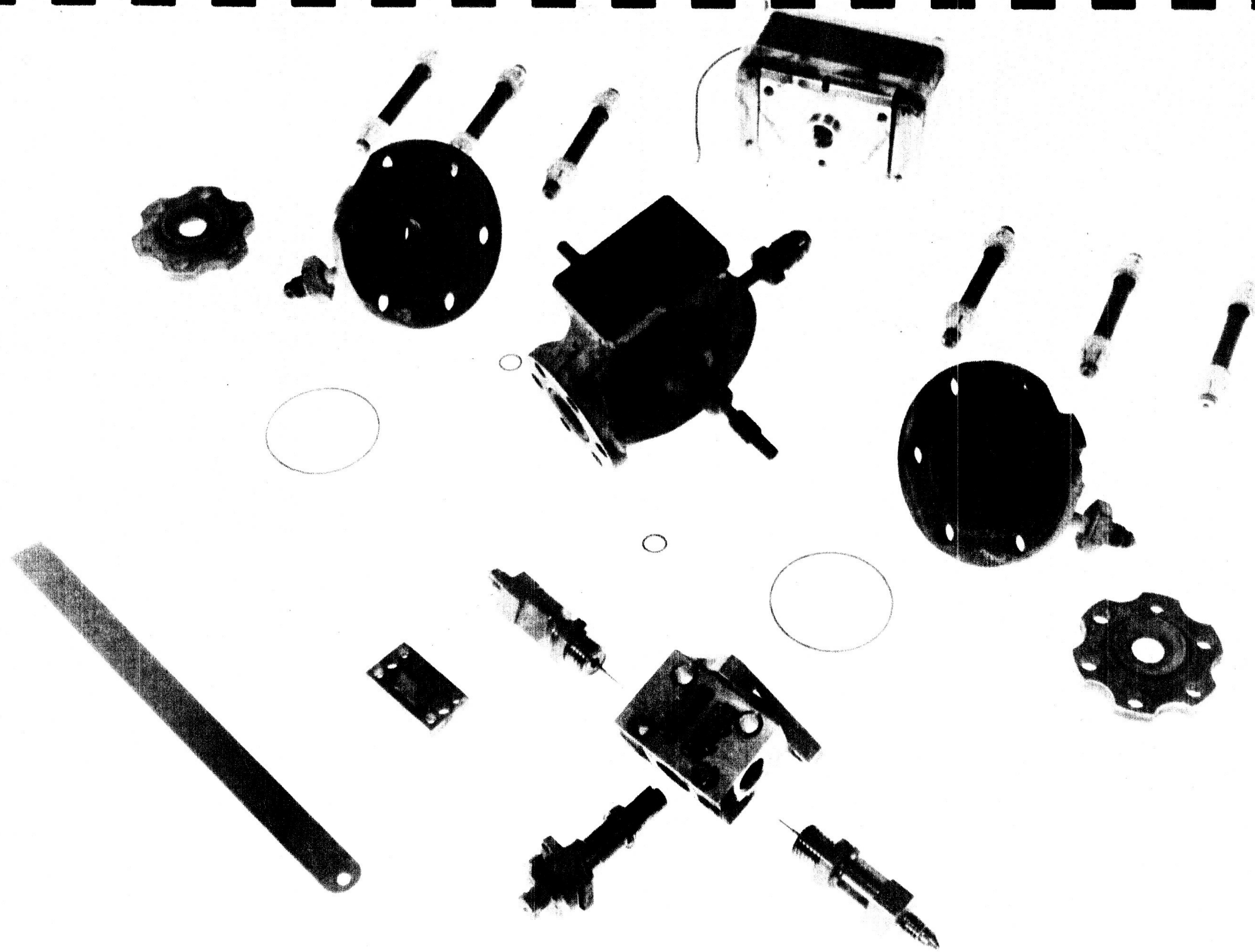
The fifth test firing of the two-stage pressure feedback valve proved successful. The structural integrity of the redesigned support shell, which encases the graphite bushings and the redesigned end cap discharge flanges was proven at operating temperatures. Figure 16 illustrates the structural integrity of the flightweight (4.24 lb.) hardware.

Excellent experimental dynamic response data was recorded for approximately 90% of the 43 second duration test. Null biasing was observed during the final five seconds of the hot test, which precluded acceptable valve performance. In addition, moderate output sine wave distortion was observed at the high frequencies (20 to 40 cps) during test.

Post examination and mathematical analysis of the servo section thermal expansion showed that the distortion and biasing were caused by a marginal "lock down" condition of the servo section at operating temperature. The sine wave distortion was reproduced experimentally under gaseous nitrogen conditions by simulating a loose servo section. The projected solution was to incorporate belleville springs on the first stage to insure holddown of the servo at temperature and to minimize null biasing.

2000°F Test #6

This test was conducted to determine the system dynamic response performance. The sixth unit was successfully test fired for 46 seconds duration. Excellent dynamic and frequency response data were recorded throughout the 2000°F pneumatic test. Null biasing, which was experienced on the fifth high temperature test, was minimized by incorporation of the belleville spring



TWO STAGE PNEUMATIC SERVO VALVE

POST FIRING

FIGURE - 16

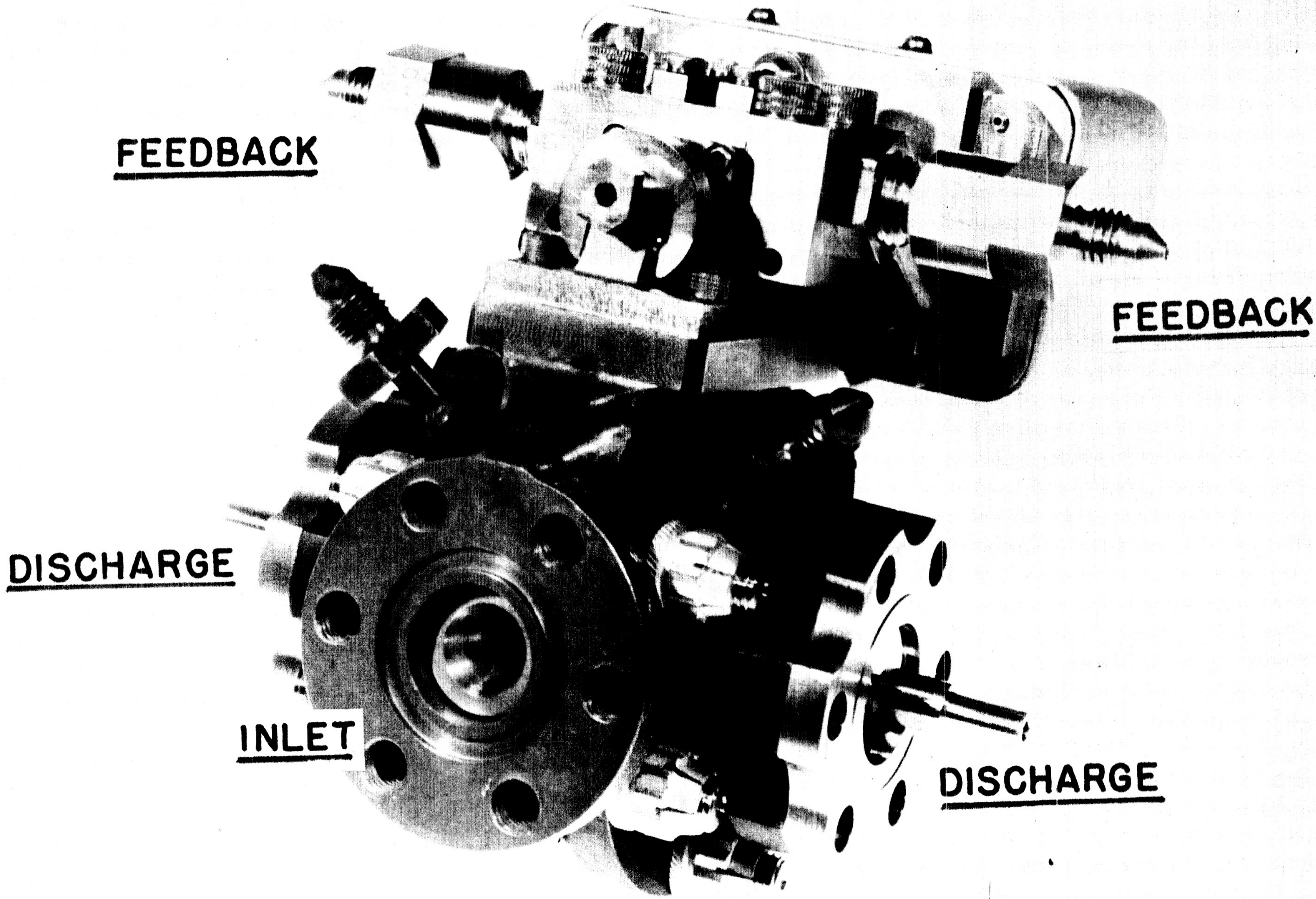
lock down on the servo stage. Closed loop frequency response was conducted out to 50 cps. This data verified the previously reported frequency response valve capability of 45 cps at 3 db attenation. A close-up view of the finalized two-stage valve hardware is shown in Figure 17.

4.3.1 Static and Dynamic Performance

Experimental tests conducted on the two-stage valve under both gaseous nitrogen and 2000°F pneumatic conditions provided an excellent correlation between 70°F and 2000°F pneumatics. Figure 18 illustrates both this correlation and the proportionality of the two stage unit under closed loop performance conditions.

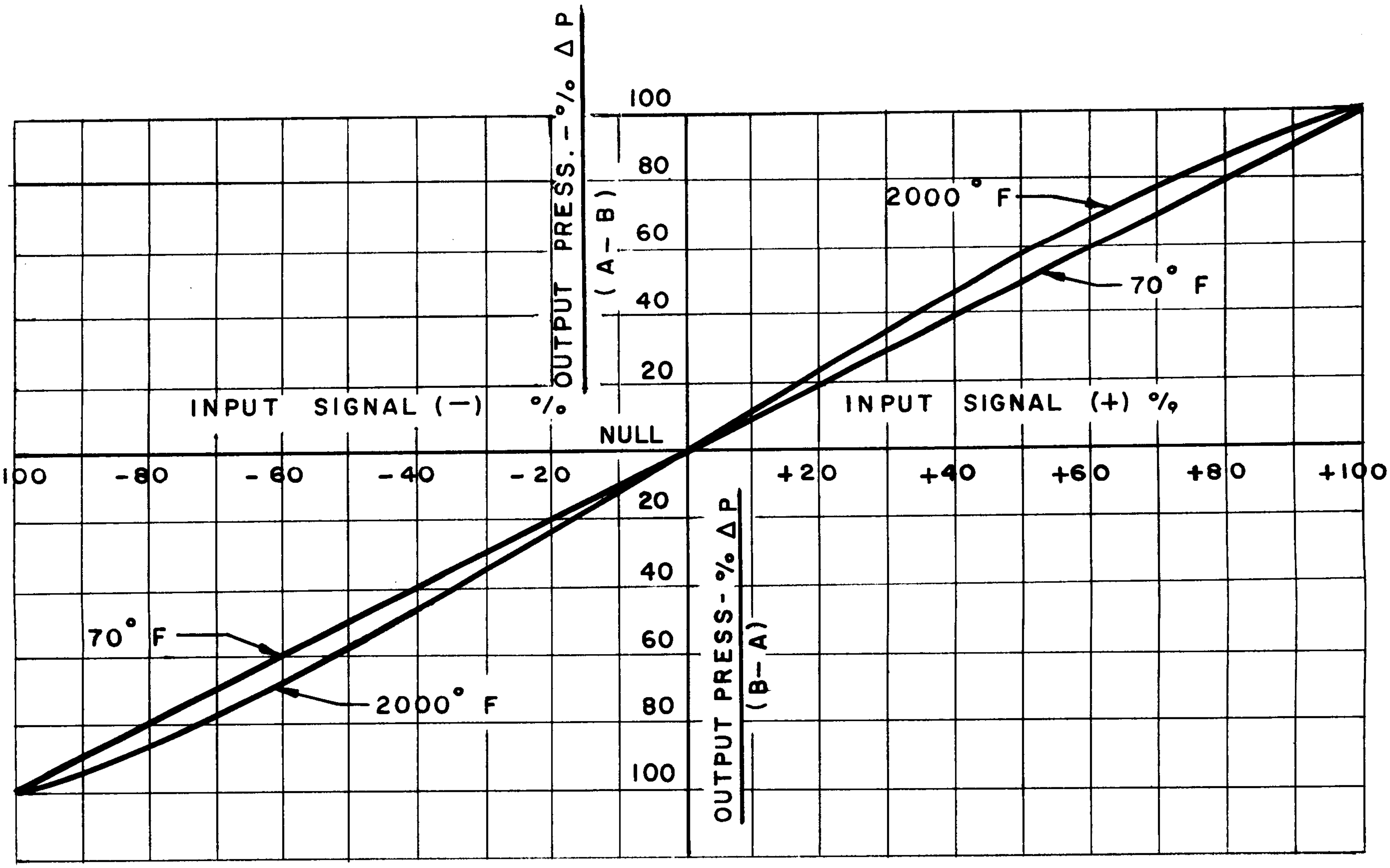
Figure 19 illustrates typical sine wave input-output performance. All frequency response data was reduced and corrected for pressure and input amplitude, and is shown in Figures 20 and 21. It should be noted that the input signal amplitude was approximately 57% and the damping ratio showed a slight increase from 0.45 to 0.5 between 70°F nitrogen and 2000°F pneumatic data.

Figure 22 illustrates typical step response traces at 2000°F for a 27% command, 56% command, and 100% command. Since it is difficult to experimentally achieve a pure step input, all experimental response times have been treated as a high rise ramp input. Therefore, response time is defined as the time



TWO STAGE PNEUMATIC SERVO VALVE

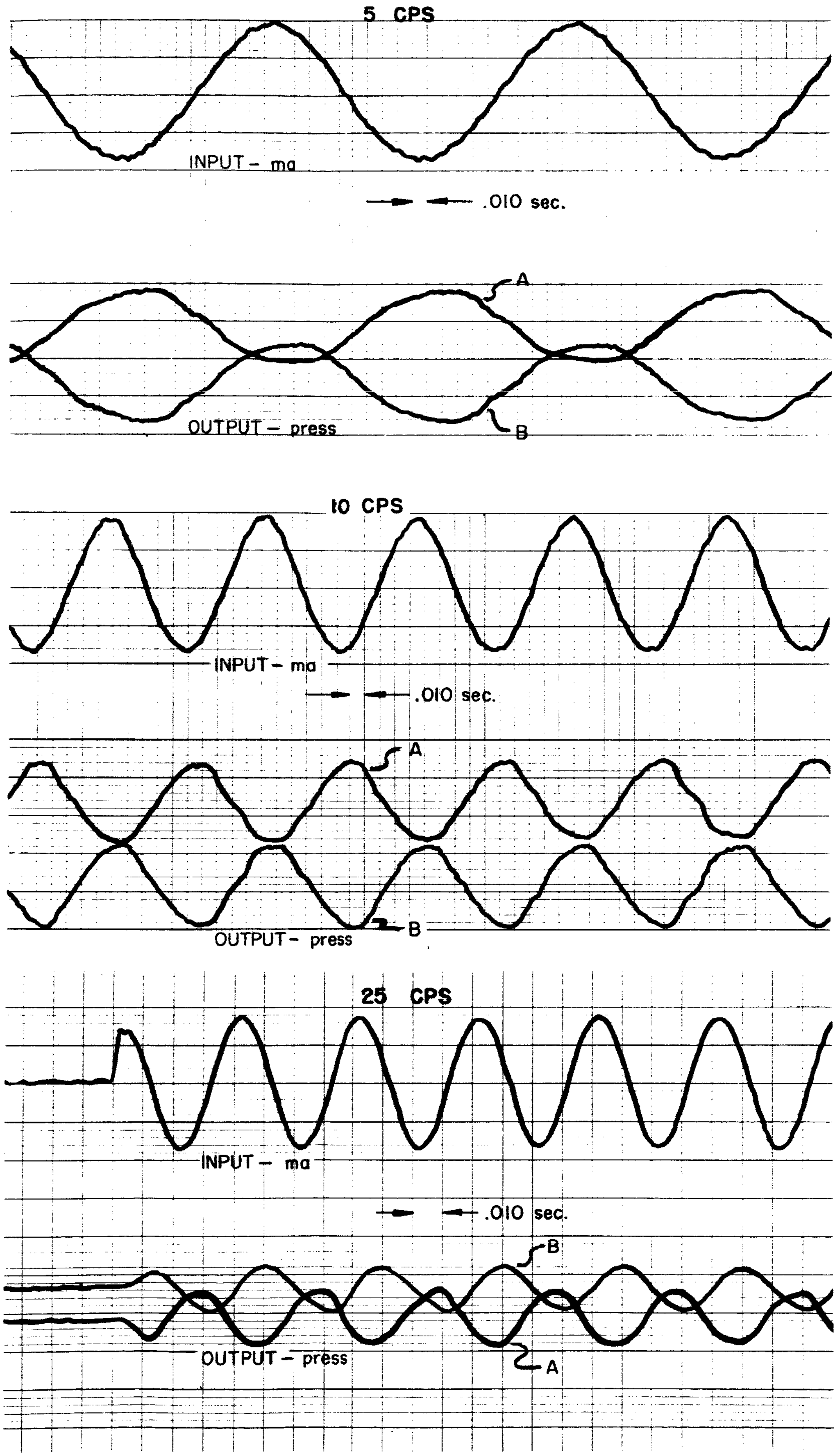
FIGURE – 17



PROPORTIONAL INPUT OUTPUT CHARACTERISTICS

TWO STAGE PNEUMATIC SERVO VALVE

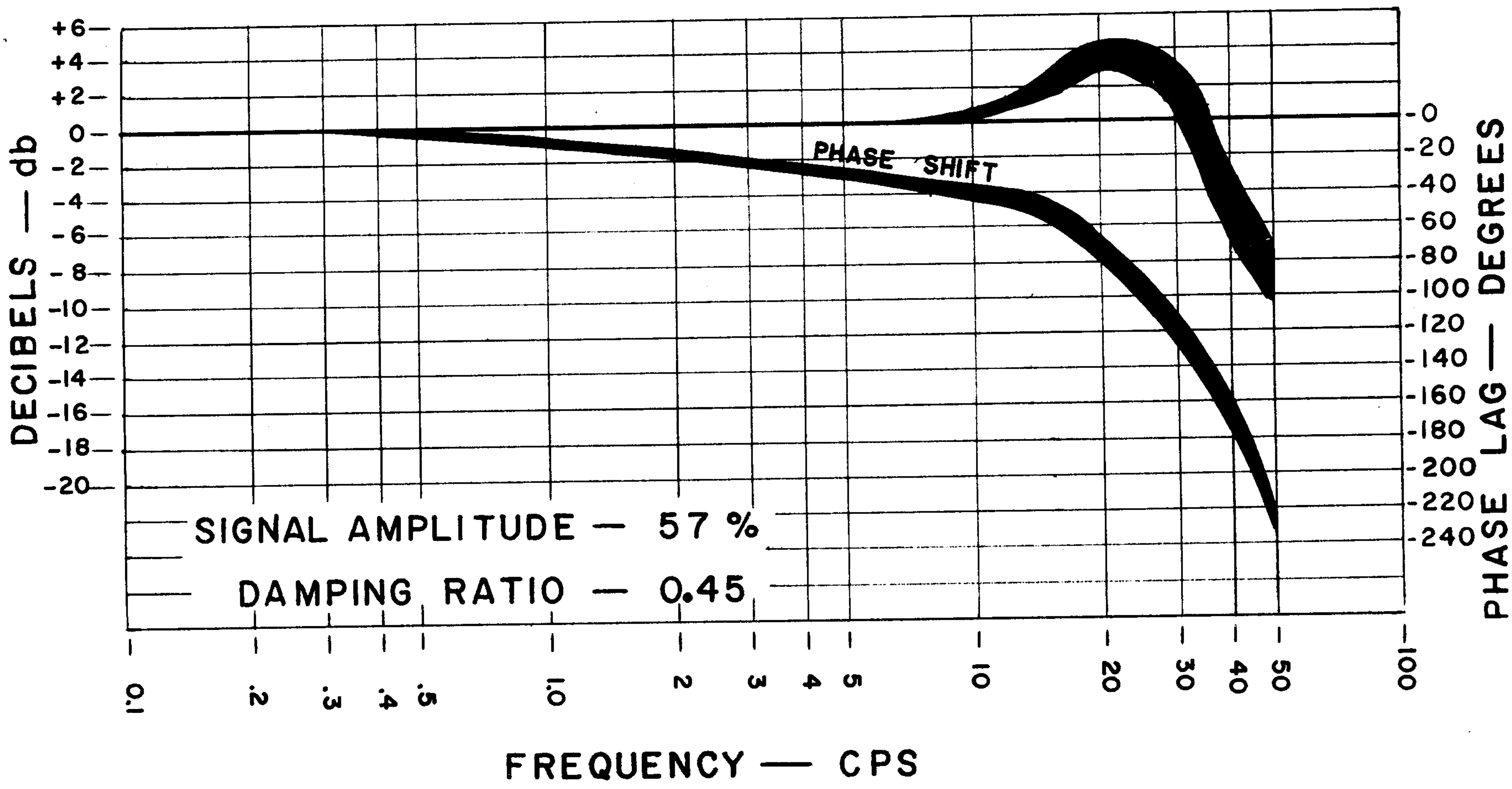
FIGURE- 18



TYPICAL SINE WAVE INPUT OUTPUT

2000° F

FIGURE - 19

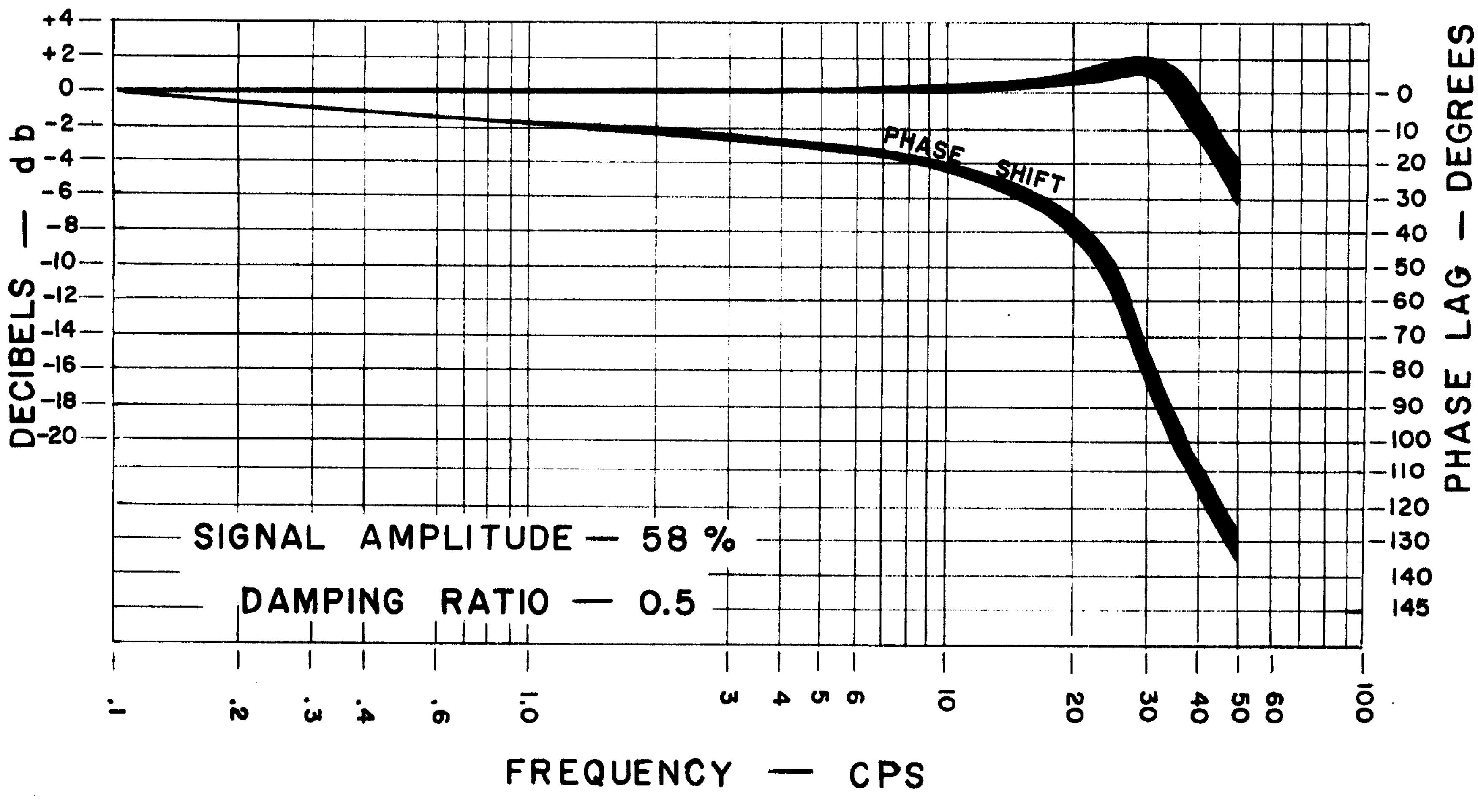


CLOSED LOOP FREQUENCY RESPONSE & PHASE LAG

TWO STAGE PROPORTIONAL PNEUMATIC VALVE

GASEOUS NITROGEN PERFORMANCE

FIGURE - 20

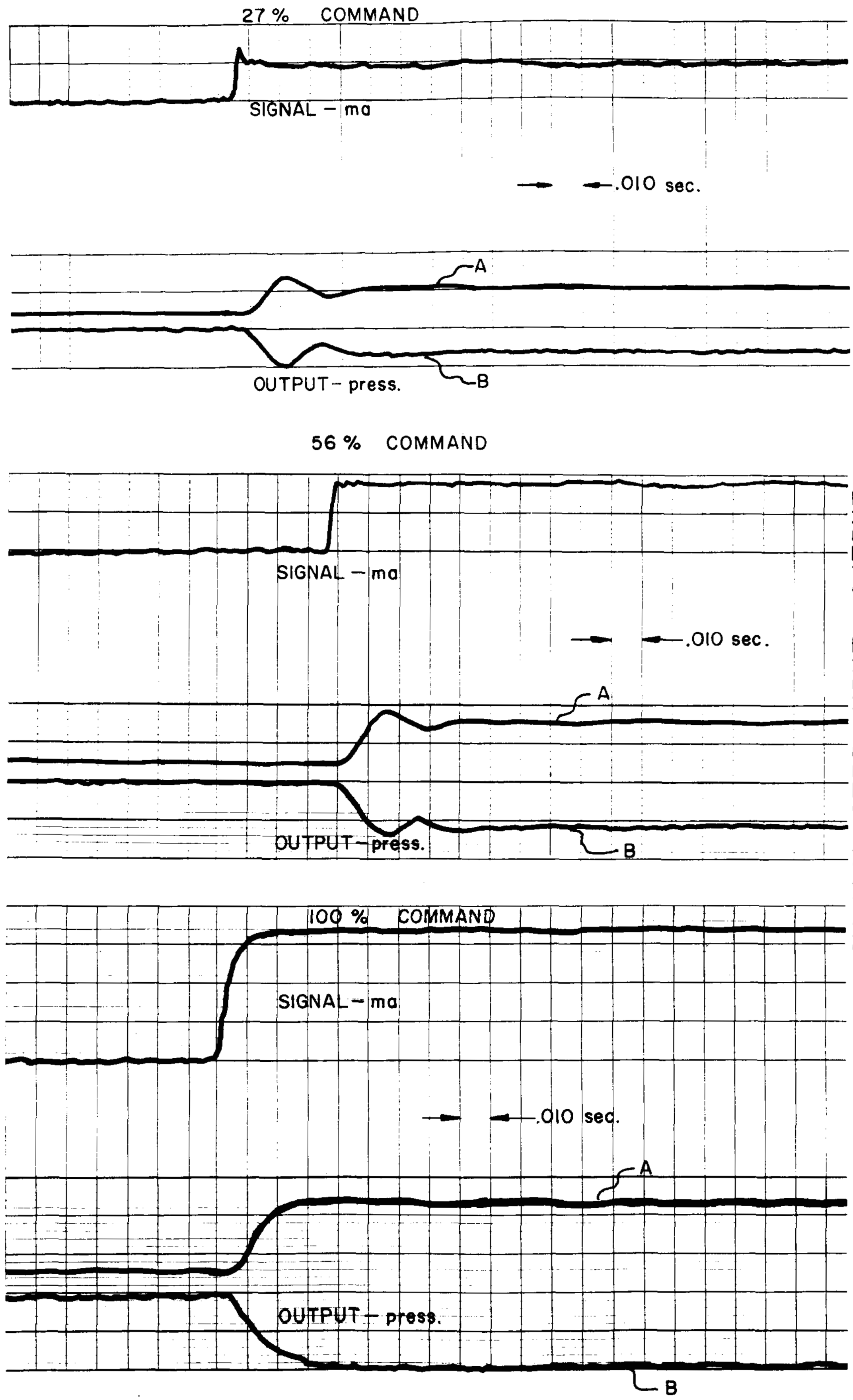


CLOSED LOOP FREQUENCY RESPONSE & PHASE LAG

TWO STAGE PROPORTIONAL PNEUMATIC VALVE

2000° F. PNEUMATIC PERFORMANCE

FIGURE - 21



TYPICAL STEP INPUT OUTPUT

2000° F

FIGURE - 22

delay between the application of an electrical input "step" signal and the resulting output "step" signal. The time measurements for the "step" signals are made at the 50% of final value point.

A summary of the dynamic response characteristics of the two-stage pressure feedback valve under 2000°F and gaseous environments is tabulated in Figure 23. For direct comparison purposes, total response time is presented as the combination of "delay" and "slew". "Delay time" is the time from initiation of electrical command to start of valve movement. "Slew time" is the time from start of chamber pressure rise to 90% of final value or first crossover assuming a second order output.

Figure 24 is a composite curve illustrating the temperature profile of the two-stage valve. As shown, the two legs of the discharge pressure are near final SITVC temperature ten seconds after ignition. The differential temperature across the valve is approximately 75°F. By using the pyrolytic graphite heat shield on the servo housing the torque motor exterior temperature never exceeded 250°F during the full test duration.

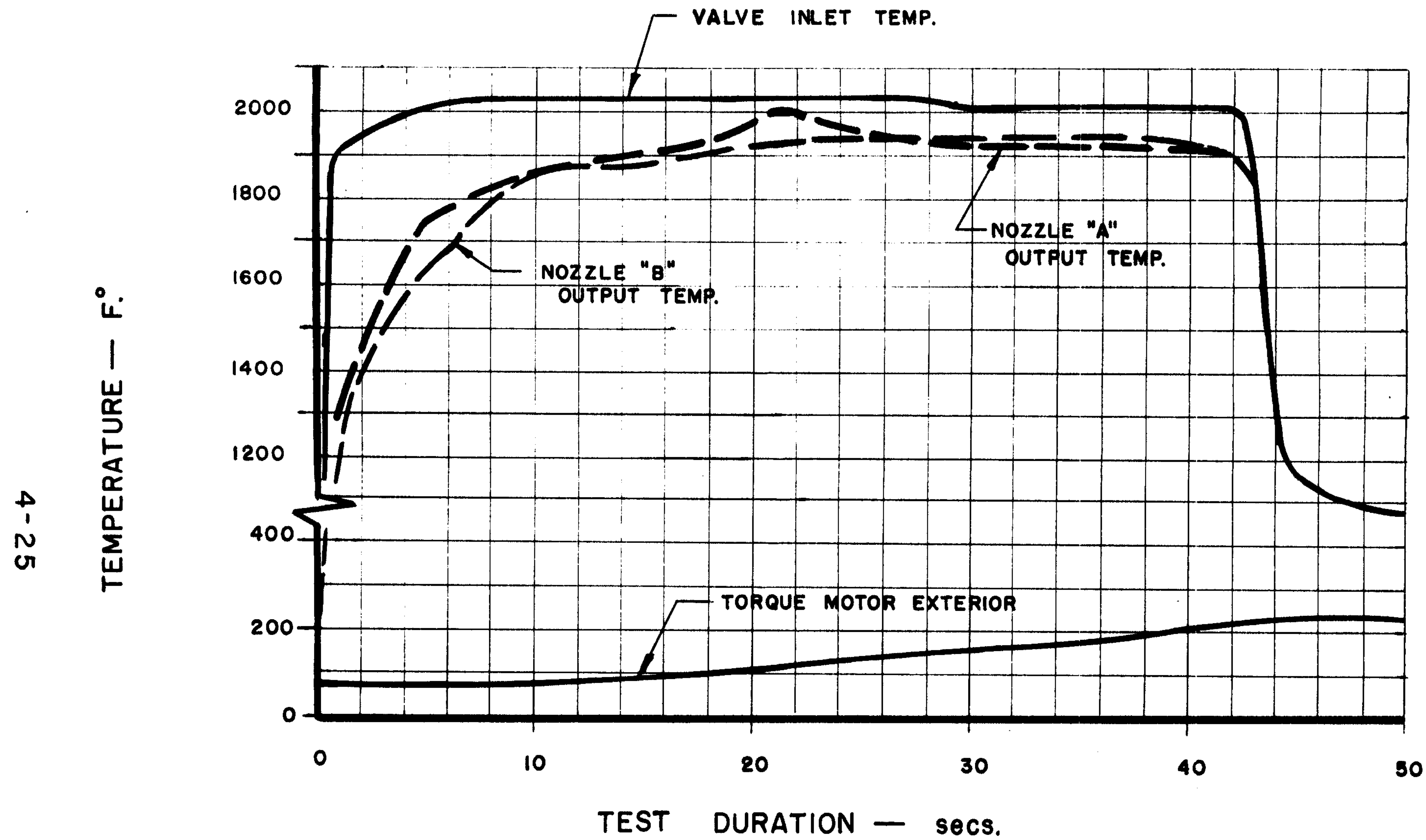
4.4 Analog - Experimental (Gaseous N_2 and 2000°F) Correlation

The correlation of valve performance when operating with gaseous nitrogen and 2000°F pneumatics is of prime importance for valve calibration and design purposes. This correlation can be

Command %	Valve Travel Mode	2000°F M-sec. Delay	Slew	% Overshoot	Gaseous N_2 M-sec. Delay	Slew	% Overshoot
6	"B" to Null	7	12	0	-	-	-
12	Null to "A"	7	11	0			
12	"B" to Null	6	10	0			
27	Null to "A"	4	4	18	7	10	0
27	"B" to Null	4	6	16	7	12	0
60	"B" to Null	4	5	30	7	9	15
60	Null to "B"	4	5	27	8	10	0
60	Null to "A"	4	6	22	7	9	0
76	"B" to Null	4	6	nil	8	7	nil
76	"A" to Null	4	5.5	nil	7	9	nil
76	Null to "A"	4	5.5	10	7	10	3
94	"B" to Null	4.5	5.5	nil	7	10	nil
94	Null to "B"	4	6	nil	7	9	0
100	"A" to Null	4.5	5	10	8	9	2.5
100	"B" to Null	4	6	8	-	-	-
+100 to -100	A to B	6	6	0	7	13	0

TWO-STAGE VALVE - EXPERIMENTAL DYNAMIC RESPONSE TABULATION

FIGURE 23



TYPICAL TEMPERATURE PROFILE
TWO STAGE PNEUMATIC SERVO VALVE

FIGURE - 24

analytically evaluated based on pneumatic flow through a sonic orifice.

The basic equation for sonic flow through an orifice is:

$$(1) \qquad \dot{\omega} = C_1 \, C_d \, A \, \frac{P_u}{\sqrt{T_u}} \qquad \text{(Sonic)}$$

where: $\dot{\omega}$ = flow rate - lb/sec

C_1 = orifice flow coefficient ($\sqrt{°R}$/sec) (dependent on the gaseous thermodynamic properties)

C_d = orifice discharge coefficient

A = orifice effective area - $inch^2$

P_u = upstream absolute pressure - psia

T_u = upstream stagnation temperature - °R

The detailed thermodynamic equation for the orifice flow coefficient (C_1) is contained in Appendix A.

C_1 for N_2 is 0.522 $\sqrt{°R}$/sec

C_1 for OMAX 453D solid propellant is 0.413 $\sqrt{°R}$/sec

Since = C_d, A, and P_u are basically constants which are not dependent on thermodynamic properties, equation (1) can be rearranged and equated from Nitrogen to OMAX 453D.

$$\dot{\omega}\,(N_2) \; \frac{\sqrt{T}\,(N_2)}{C_1\,(N_2)} = \frac{\omega\,(453D)\;\sqrt{T}\,(453D)}{C_1\,(453D)}$$

rearranging and substituting,

$$\dot{\omega}\, N_2 = \dot{\omega}\,(453D)\ \frac{0.522\sqrt{2460}}{0.413\sqrt{530}}$$

thus,

$$\dot{\omega}\,(N_2) = 2.72\ \dot{\omega}\,(453D)$$

The correlation data for the analog, experimental gaseous nitrogen, and experimental 2000°F data is tabulated in Figure 25.

It should be noted that the response times stated do not include the delay times from initiation of electrical command to start of the rise time. In addition, the experimental data is based on a servo volume of approximately double the theoretical simulated volume due to the use of instrumentation probes. The double volume would produce an increase in response since more volume demands an increase in compressibility flow.

Based on Figure 25, good theoretical to experimental correlation was achieved in respect to response, system damping ratio, and system gains. As evidenced by the data and Figure 18, a slight gain increase is experienced from gaseous nitrogen to 2000°F pneumatic performance.

4.5 Dynamic Seals

The application of special design high temperature piston rings with graphite bores for the 2000°F pneumatic environment provided

Description	Theoretical	Experimental	
	Analog	OMAX 453D	Gaseous N_2
	2000°F	2000°F	70°F
A. *Step Response-50% command amplitude			
Rise Time -M-sec	4.0	6-8	9 to 11
% Overshoot	33%	10 to 18%	15 to 28%
Settling Time -M-sec to 90% of Final Valve	12	9 to 14	nil
B. Damping Ratio	0.40	0.50	0.45
C. System Gain $\left(\frac{\Delta \text{ Output}}{\Delta \text{ Input}} = \frac{\text{PSI}}{\text{MA}} \right)$	14.4	15.0	13.7

* Based on the assumption that dynamic performance approximates a second order system.

CORRELATION OF THEORETICAL AND EXPERIMENTAL

FIGURE 25

an excellent dynamic seal configuration. As shown in Figure 1 (Section 2) dual piston rings were used on each spool diameter for reliability purposes. The servo and pneumatic rate effective spool areas were diametrically sealed with a combination two ring seal. The outer ring was fabricated from a ductile iron-parco lubrized alloy and the inner expander ring was fabricated from Inconel X material. The center shaft incorporated only the outer ductile iron-parco lubrized alloy piston ring. Use of the inner expander ring on the center shaft was not permitted due to size limitations. The Stackpole 331 graphite exhibited moderate wear after N_2 calibration tests of five to ten minutes duration but did not affect sealing performance. All dynamic seals performed according to design expectations and are considered acceptable and reliable for 2000°F pneumatic environments.

In general, the gold plated Inconel X metal "Vee" static seals performed satisfactorily for purposes of the development program. However, they are not considered reliable since seal leakage was experienced unpredictably regardless of the extremely close mechanical tolerances and finishes required. The pneumatic leakages experienced during the 2000°F testing were tolerable and did not appear to affect valve performance for the 43 second nominal duration test.

The Raybestos Manhattan A-56 gasket static seals were used only at the interface of the valve and discharge manifolds. The gasket material performed satisfactorily throughout the test duration.

4.6 Performance of Construction Materials

BG-42 Vac Arc (Latrobe Steel Company) high speed steel was used for the basic valve housings. BG-42 is essentially a martensitic stainless steel. The valve housings were fabricated using the BG-42 material in the annealed condition with the exception of the first test unit.

As a result of the 2000°F pneumatic environment tests, three undesirable material characteristics were experienced.

(1) BG-42 material is thermal shock sensitive when used in the heat treated condition.

(2) When the annealed condition BG-42 material valve was subjected to 2000°F for the 43 second duration testing, the internal valve sections were heat treated non-uniformly to hardness values in the Rockwell "C" 58 range. Since the internal sections were essentially "heat treated" and the external case was annealed, the valve case was subjected to extremely high internal stresses. As a result the valve housing would crack in a 2 to 15 hour period following hot test.

(3) A high degree of "creep" at temperature was experienced using the BG-42 material. For example, the inlet sonic

orifice (fabricated from RA-330) which exhibits a higher coefficient of expansion than BG-42 was assembled with a press fit in the valve housing. Following a test firing the inlet orifice was completely free and would dislodge from the valve with a minimum of effort. In addition, there is evidence of leakage around the inlet orifice during one test firing.

It should be noted that the BG-42 exhibited a complete loss of corrosion resistance following a test firing and showed average corrosion resistance in the annealed state prior to test firing.

The spool and metering orifices (including servo orifices) were fabricated from RA-330, an austenitic super alloy stainless steel which exhibits excellent stress rupture and creep properties for 2000°F service. In all test firings, RA-330 performed very satisfactorily and did not exhibit any dimensional change. This material was subjected to sonic and supersonic 2000°F pneumatic flows in the metering section of the valve and exhibited excellent erosion resistance.

No evidence of deterioration was experienced using the Stackpole 331 graphite bores. The operational performance was satisfactory.

The pyrolytic graphite heat shield which served as a thermal barrier for the torque-motor-servo section from the main valve housing was used for all six (6) 2000°F tests. No evidence of deterioration was observed.

All miscellaneous materials used in other valve areas, 303 stainless, A-286 studs and lock nuts, AM-350 bellows, and molybdenum (M_o+1/2T_i) yoke and shaft performed to design specifications.

SECTION 5

INVESTIGATION OF TWO-STAGE VALVE 5500°F POTENTIAL

5.1 Effect of Major Parameter Variations

The two-stage valve design concept was explored to determine the effect of 1500, 2000, and 2700 psi system supply operating pressure levels, mass flows of 0.58 to 5.0 lb/sec, and operation at 2000°F and 5500°F pneumatic supplies.

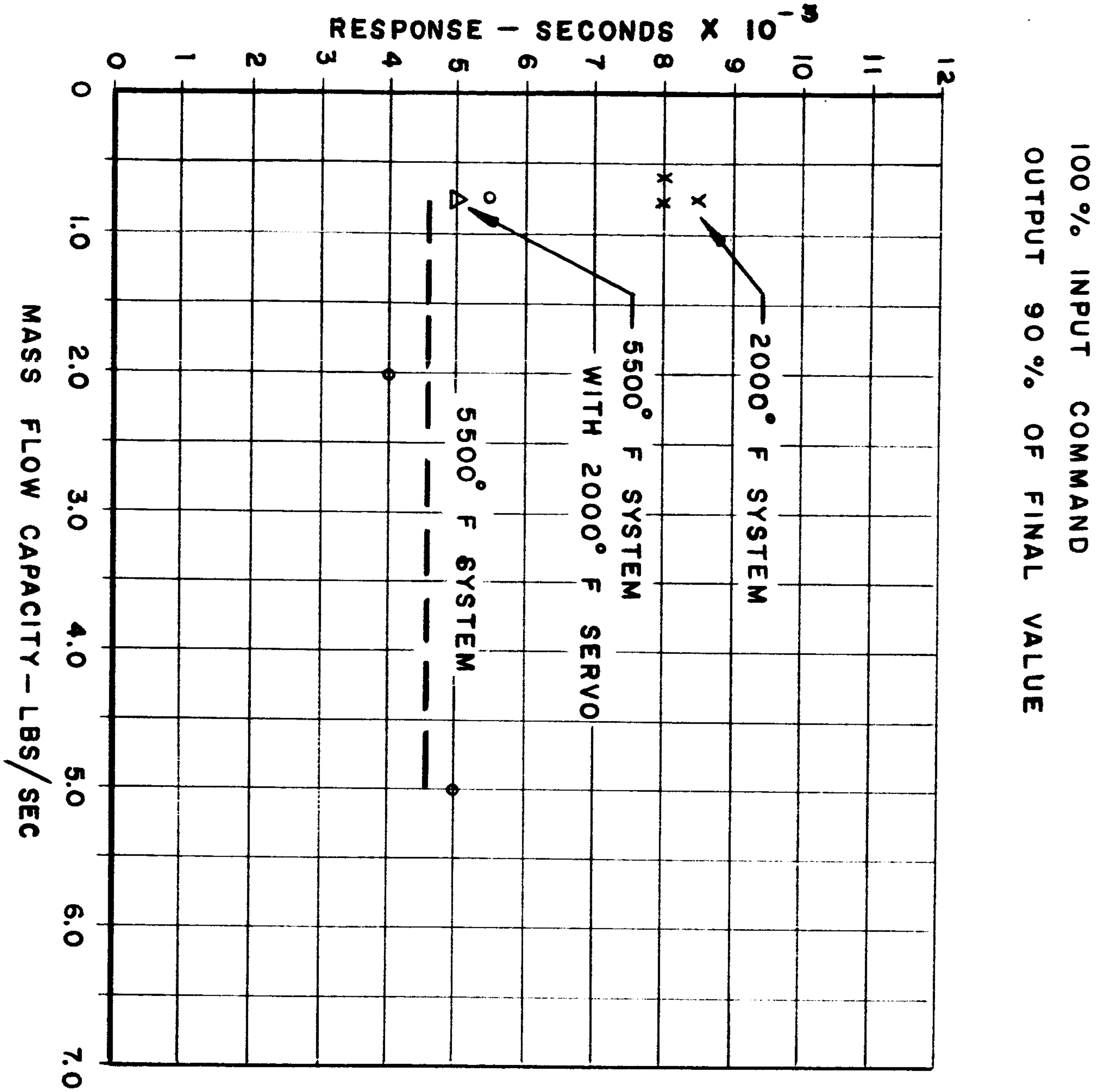
The results of these studies are tabulated in Figure 26. Considering the pressure variation range studies, (1500 to 2700 psi) no specific improvement could be extrapolated from the data for the higher pressure system. A definite improvement in dynamic response can be realized by increasing the pneumatic fluid operating temperature level from 2000°F to 5500°F.

Figure 27 illustrates the magnitude of improvement. As shown, the 2000°F system provides a response in the order of 8 milliseconds. The 5500°F system provides a response in the order of 4½ milliseconds. It should be noted that a two-stage valve design using a 5500°F primary flow stage with a 2000°F servo flow stage provides a dynamic response capability equivalent to a 100% 5500°F pneumatic system. This would indicate that the main contribution affecting dynamic response is the downstream discharge manifold volume rather than any dynamic limitations

	VALVE PARAMETER, SIZE, AND CHARACTERISTICS	VH-20-L	H-20-L	M-20-L	M-55/20-L	M-55-L	M-55-M	M-55-H	H-55-L
1	Inlet or generator pressure - psia	2715	2015	1515	1515	1515	1515	1515	2015
2	Valve inlet (and servo) temperature - °F	2000	2000	2000	5500 (2000 servo)	5500	5500	5500	5500
3	Total mass flow/servo size - lb/sec	0.58/4.0%	0.704/3.27%	0.704/3.27%	0.704/3.16%	0.704/3.27%	2.0/4.0%	5.0/4.0%	0.704/3.27%
4	Servo leakage - % of servo flow	7%	7%	7%	7%	7%	7%	7%	7%
5	Main stage stiction/friction-lbs	13/10	60/50	86/72	86/72	86/72	172/144	243/203	60/50
6	Valve pneumatic spring rate - lb/in	3640	2930	2860	2800	2840	2710	2680	1920
7	Output or discharge volume (one side) $inch^3$	9.42	9.42	9.42	9.42	9.42	22.5	58	9.42
8	Electrical power consumption - watts	4.1	5.5	6.0	6.0	6.0	-	-	5.5
9	Valve weight - lbs	4.0	4.24	4	7.4	9.3	19.8	34.0	9.3
10	Dynamic response:								
	High temperature - step to 1st crossover or 90% final valve								
	25% command - % overshoot - settling time-m-sec.	7/25%/8	6.5/0/0	7.5/0/0	6.5/0/0	4.5/0/0	-	-	5/0/0
	50% command - % overshoot - settling time-m-sec.	5.0/26.5%/10	5.5/26.2%/9	6/12.5%/5	5/0/0	4/0/0	-	-	4/20%/.5
	75% command - % overshoot - settling time-m-sec.	7/13%/7	8.5/8.9%/9	7.5/0/0	6.5/0/0	4.5/0/0	5/0/0	-	-
	100% command - % overshoot - settling time-m-sec.	8/0/0	8.5/0/0	8/0/0	5/0/0	5/0/0	4/0/0	5/0/0	5.5/6.25%/7

MAJOR PARAMETER VARIATION RESULTS

FIGURE 26



EFFECT OF SYSTEM TEMPERATURE LEVEL ON STEP INPUT DYNAMIC PERFORMANCE VS. MASS FLOW LEVEL

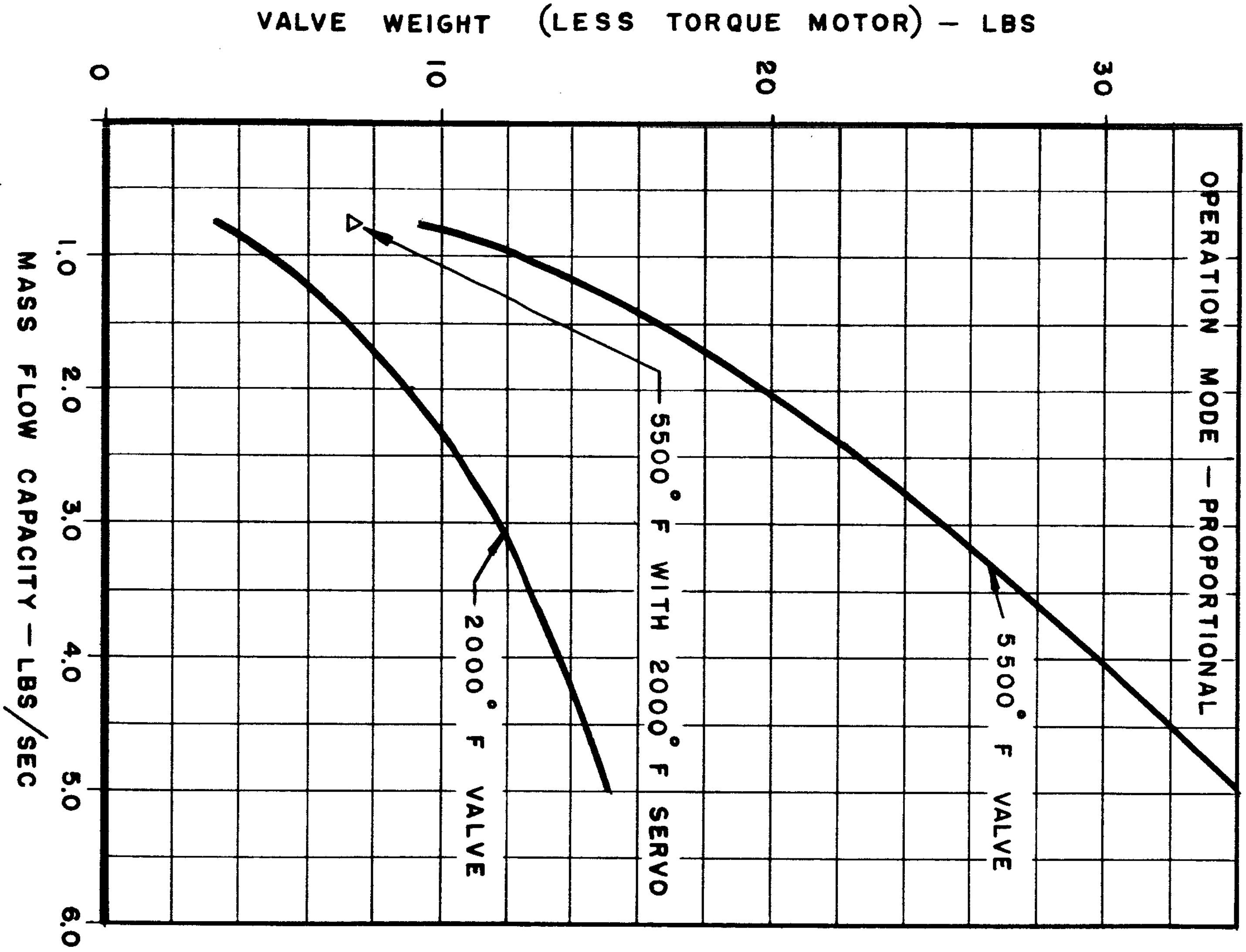
FIGURE - 27

from the servo stage. Figure 27 also illustrates that specific response capability can be achieved independent of mass flow capacity by selective design of the servo stage.

Figure 28 illustrates the scaling of the two-stage valve with respect to mass flow capacity for a 2000°F and 5500°F system. The main contribution for weight increase for 5500°F system operation is the materials integrity for 5500°F service. Obviously a trade off point will be realized where electrical torque motor staged performance will yield to hydraulic or more practical means of control for the higher mass flow capacities.

5.2 5500°F Two-Stage Conceptual Design

The two-stage electro-pneumatic-mechanical design concept was APPLIED to explore the design concept potential for use in 5500°F aluminized propellant systems. Based on this study, Figure 29 shows the conceptual design of a two-stage proportional 5500°F pneumatic valve. Although, alternative designs such as using a 2000°F servo amplification stage could be used for controlling the 5500°F primary flow, this study was based on using a percentage of the 5500°F inlet gas for servo control. As shown in Figure 29, the basic construction is comprised of ablative materials and tungsten in the critical flow metering areas. The main spool is "buried" in an ablative insulation structure to minimize thermal heating of the main spool section. The main spool

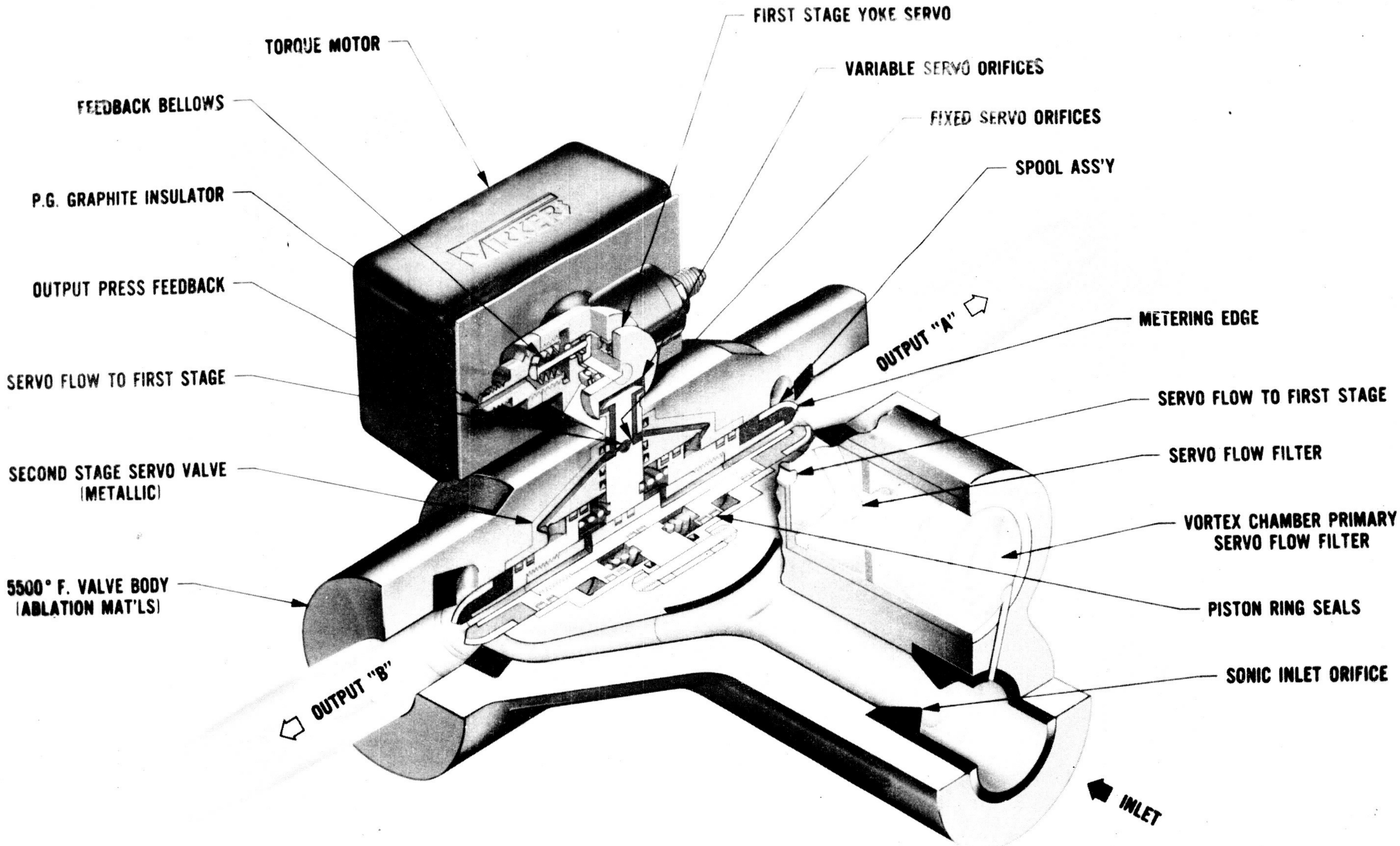


TWO STAGE VALVE (LESS TORQUE MOTOR)

FLIGHT WEIGHT VS. SCALED MASS FLOW

2000° F AND 5500° F SYSTEMS

FIGURE - 28



TWO STAGE 5500° F. PNEUMATIC PRESSURE FEEDBACK VALVE

FIGURE - 29

is a multi-material construction to utilize tungsten for the metering edge backed by an insulator and super alloy stainless for the "buried" spool control sections. Tungsten materials are recommended for the servo yoke and orifice section. The torque motor is protected by a grain oriented pyrolytic graphite shield.

The bellows feedback mechanisms would utilize AM-350 or Inconel "X" material bellows based on previous verification tests in a 5600°F pneumatic environment. The application of stainless bellows materials in 5600°F environments is functionally achieved by utilizing the low thermal conductivity stagnated gas barrier principle such as was applied to the developed 2000°F two-stage valve.

With the exception of materials and construction, the design concept and system logic is identical to the developed two-stage valve. The system logic is explained in detail in Section 3.

Since the 5500°F valve concept is designed to ingest the combustion products of alumized propellant, the servo must be protected from plating of the aluminum oxide and plugging of the servo orifices. This is accomplished by incorporating a combination servo filter. For this study, the following design and propellant formulation requirements were assumed.

Propellant Ballistic Data:

Arcite 373 Gel Type (Atlantic Research Corp.) composition

% by weight.

Ammonium Perchlorate -	58.90
Aluminum	21.10
Polyvingl Chloride	8.62
Dioctyl Adipate	11.13
Wetting Agent	0.25

Combustion Products (% by weight):

CO	25.51
CO_2	0.35
H_2	38.30
H_2O	4.00
HCL	13.41
N_2	6.81
Al Cl	2.45
Al_2O_3 (liquid)	9.17
Al_2O_3 (solid)	0.00

Thermodynamic Properties:

Flame Temperature - T	3317°K
Ratio of Specific Heat - K	1.185
Molecular Wt. - M	29.5
Gas Constant - R	52.34 ft/°R

5500°F Two-Stage Valve Design:

Flow Rate - $\dot{w}$ - 0.704 lb/sec
Servo Flow Rate - 3% of .704 = .0211 lb/sec
Propellant Combustion Products = 9.17 wt % Al_2O_3

Assuming a 60 second duration test

Total mass flow = .704 lb/sec x 60 sec = 42.24 lb
Total servo flow = .0211 lb/sec x 60 sec = 1.264 lb

Therefore: 9.17% of 1.264 lb. servo flow

= 0.116 lb. alum to servo

The density of Al_2O_3 is .128 lb/in^3

$$\therefore \quad \frac{0.116 \text{ lbs}}{.128 \text{ lbs/in}^3} = .909 \text{ in}^3 \text{ of aluminum to servo}$$

Assuming that the density of the aluminum oxide product is approximately one-third its normal density when it is at 5500°F temperature, then the total aluminum oxide ingested by the servo would be equivalent to 2.7 cubic inches during a 60 second duration. As shown in Figure 29, the 5500°F valve incorporates a vortex chamber primary filter and a porosity type secondary bypass filter. The vortex filter is intended to remove 90% of the aluminum oxide and the secondary filter is intended to approach 100% filtration.

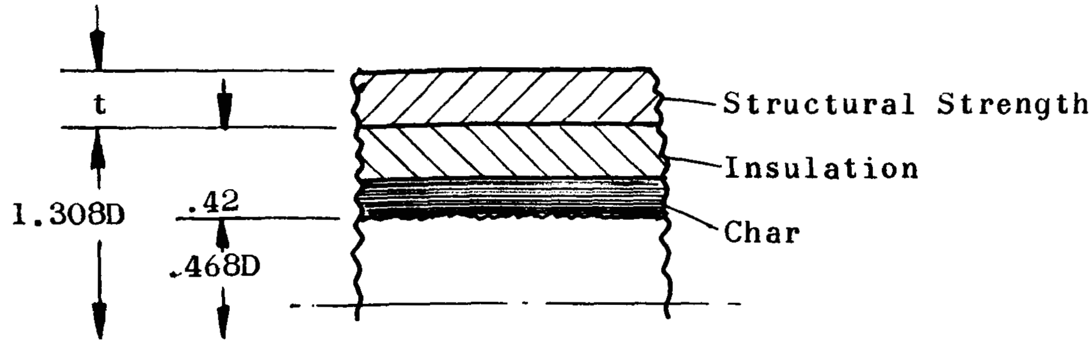
5.2.1 Materials and Stress Analysis

For the purpose of direct comparison with the 2000°F developed two-stage valve, the mass flow capacity and the system operating pressure levels were assumed at .704 lb/sec and 2000 psi supply pressure. A survey of candidate materials for the 5500°F valve was conducted and a graphite laminate, designated CCA1 was selected for the ablative valve housing construction. The following material - stress analysis was conducted to determine the valve sizes.

5.3 Ablative Valve Body Strength

Valve Body

CCAI - Ablative Material Hitco Gardena, California



Paragraph 4 of Test Firing Data for ablative performance states *1

"Average erosion rate is 7.0 mils/sec" .007 x 60 = .42 inch

The .468D is equivalent to the area of pneumatic flow slots in the end cap of 2000 psi valve #100051-x

<u>Hoop Stress</u>: $S_2 = \frac{PR}{t}$ Roach p268 Case 1 *2

Where: S = Stress - psi

P = Internal Press - psi

R = Radius - Inch

t = Wall Thickness - Inch

<u>Thru Passage</u>

p = 750 psi

let t = .300 R = .654 + .15 = .804

$$S_2 = \frac{(750)(.804)}{13} = \frac{603}{.3} = 2010 \text{ psi}$$

Tensile strength Virgin Laminate = 20,200 *1 10:1 S.F.

Tensile strength Charred Laminate = 7,400

---

*1 Millington, R.B., Carbon & Graphite Ablative Reinforcements, Hitco, Feb., 1965.

*2 Roark, Raymond J., Formulas for Stress & Strain, McGraw Hill, New York, 1954.

$\therefore$ Valve body diameter in area of flow area is to be

$$1.308 + .6 = 1.900 \text{ inch}$$

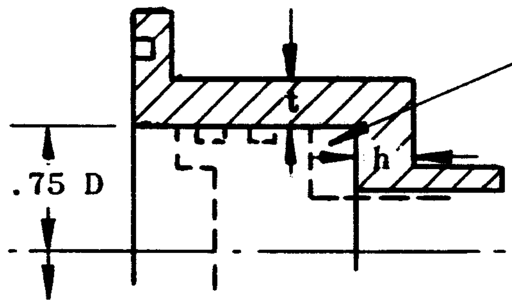
At Outlet

p = 375 psi let S = 4000 psi Find t R - 1.0 inch

$$4000 = \frac{(375)(1.0+t)}{t} \qquad t = .103$$

$\therefore$ Valve body diameter at outlet is 2.00 + .2 = 2.20 inch

Second Stage (Metallic)



Hoop Stress - (S_h)

$$S_h = \frac{pR}{t} \qquad R = .375 + t/2$$

$$S_h = \frac{1500\ (.375+.5t)}{t} = \frac{562.5 + 750\ t}{t}$$

$$\text{let } t = .06 \quad S_h = \frac{562+750(.06)}{.06} = \frac{607}{.06} = 10{,}100 \text{ psi}$$

$$\text{let } t = .10 \quad S_h = \frac{562+750\ (.1)}{.1} = \frac{637}{.1} = 6370 \text{ psi}$$

Shear Area: $.75\pi\ h = 2.36h$

Area Under Press: (.4418-.1251) = .3167 in^2 @ 1500 psi

Load: .3167 (1500) = 475 lb.

Axial
Shear Stress: $S_s = \frac{P}{A} = \frac{475}{2.36h}$

let h = .125 $\quad S_s$ = 475/.295 = 1610 psi

The following weight analysis illustrates the anticipated flightweight based on selected materials capable of the 5500°F environment.

Weight Estimate

Metal portion of valve including torque motor 3.5 lb

Ablative Material of Body

Outlet	=	11.42
Body	=	25.44
Inlet	=	11.88
Passage	=	7.16

$58.87 \text{ in}^3 \times .05 \text{ lb/in}^2 = 2.9 \text{ lb.}$

Tungsten
Filter $(2.5-2.0)(2)=(4.9-3.14)(2)=3.52 \text{ in}^3 \times .688 \text{lb/in}^3 = 2.4$ lb

Filter and Baffle =1.5 lb

Total Weight 10.3 lb

SECTION 6

CONCLUSIONS AND RECOMMENDATIONS

The two-stage proportional 2000°F pneumatic valve designed and evaluated under Contract NAS 1-4102 is intended for application to flightweight solid propellant systems for either thrust vector control or attitude (PYR) control. The following conclusions are based on the program described in this report:

6.1 Conclusions

(1) The two-stage valve operationally satisfied all of the primary design and functional objectives established at the initiation of the program.

(2) The final flightweight design, which included pressure probes for instrumentation purposes, weighed 4.24 lbs. and required an electrical power of 5.5 watts.

(3) The system demonstrated proportional flow modulation throughout a range greater than 110 to 1.

(4) A system sinusoidal frequency response of 45 cps at -3 db attenuation was demonstrated in the closed loop control mode under 2000°F pneumatic conditions.

The gaseous nitrogen system sinusoidal frequency response was 38 cps at -3 db attenuation.

(5) A slight increase in output/input system gain was observed when tested at 2000°F, compared to the gaseous nitrogen tests.

(6) The system damping ratio increased from 0.45 (gaseous N_2) to 0.5 when subjected to 2000°F pneumatic testing.

(7) The typical output step response curves approximated a second order system. System step response from initiation of electrical command to output (defined in Section 4.3) ranged from 10 to 12 milliseconds using 2000°F pneumatics. Gaseous nitrogen experimental step response time was increased approximately 30% over that of the 2000°F pneumatic output response time.

(8) Good correlation between the theoretical and experimental data was demonstrated based on step response, damping ratio and system gains. This basic correlation can be applied effectively for extrapolation of performance to applications requiring different pneumatic thermodynamic characteristics.

(9) All testing was conducted on a system basis which encompassed simulation of a typical single plane SITVC system for a solid propellant motor of 2800 lbs. thrust (sea level). Therefore, the system included all required manifolds for TVC.

(10) This program has resulted in the conceptual design and

sizing of a SITVC valve for use in 5500°F aluminized solid propellant systems. The conceptual design is the result of a materials and stress evaluation for 5500°F service and the application of the developed 2000°F valve concept potential.

(11) The flightweight concept of the 5500°F two-stage valve (which is based on equivalent flow capacity of the 2000°F developed valve) would weigh 10.3 lbs (see Figure 29). This weight is based on the materials and stress levels encountered for 5500°F service.

(12) Based on the results of the analog computer study, an approximate increase of 35% in dynamic performance is realized by application of 5500°F pneumatics over 2000°F pneumatics.

For the system pressure levels investigated (1500, 2000, and 2700 psi), all dynamic responses were within an acceptable operational range of 4 to 8.5 milliseconds. In addition, these response data are based on valve capacities ranging from 0.704 lb/sec. to 5.0 lb/sec. including the associated servo and discharge manifold volumes for practical flightweight SITVC applications.

(13) An increase in two-stage valve mass flow capacity from 0.704 lb/sec. to 5.0 lb/sec. for 2000°F service would result in a flightweight of 15 lbs. An increase in two-stage valve capacity from 0.704 lb/sec. to 5.0 lb/sec. for 5500°F service

would result in a flightweight of 34 lbs. (see Figure 28).

6.2 Recommendations

The design and development of a flightweight two-stage 2000°F pneumatic pressure feedback valve for proportional SITVC systems has resulted in advancing the technology attendant to this type of system with regard to dynamic response, thermal transients, materials performance, and combustion product contamination. The analog evaluation of major parameter variations and use of analytical to experimental correlation data provides an accurate foundation for scaling this design concept for a specific application.

Three logical recommendations are presented to increase the knowledge and technology of the two-stage pneumatic valve system concept.

(1) As a result of this contract, the necessary components and performance criteria exist to design, develop, and test a flightweight system. It is recommended that a flight test program be initiated using the flightweight pressure feedback valve to control a 2000°F solid propellant proportional secondary injection system. This work would provide operational flight verification of a warm gas TVC system. The applicable SITVC parameters and results are covered in Phase I of this contract and are reported in a separate report.

(2) Based on the uprated design concept potential study conducted herein, the concept is directly applicable for application to 5500°F solid propellant rocket engine systems utilizing direct engine bleed for SITVC. It is recommended that two-stage valve development for a 5500°F aluminized propellant system be pursued for future flight systems.

(3) Since uprating of the developed valve from a 2000°F environment, using a relatively clean propellant, to 5500°F aluminized propellant presents some development problems for ingestion of aluminum oxide in the servo stage, incorporation of a 2000°F propellant for servo stage operation and 5500°F direct engine bleed for primary SITVC flow may prove more practical on a system basis. It is recommended that a system trade-off study be conducted to define this concept and based on this study, to design and develop a 5500°F valve utilizing a 2000°F servo stage.

APPENDIX A

STEADY STATE ANALYSIS FOR A FLIGHTWEIGHT 2000°F SITVC SYSTEM

This analysis provides the basic mathematical equations which were applied to size the two stage valve and to resolve the system steady state performance. Figure 30 provides a simplified schematic of the system except for the solid propellant generator.

Design Parameters

The system design parameters for both 2000°F and gaseous nitrogen pneumatics is shown in Figure 31. This analysis is based on the following assumptions:

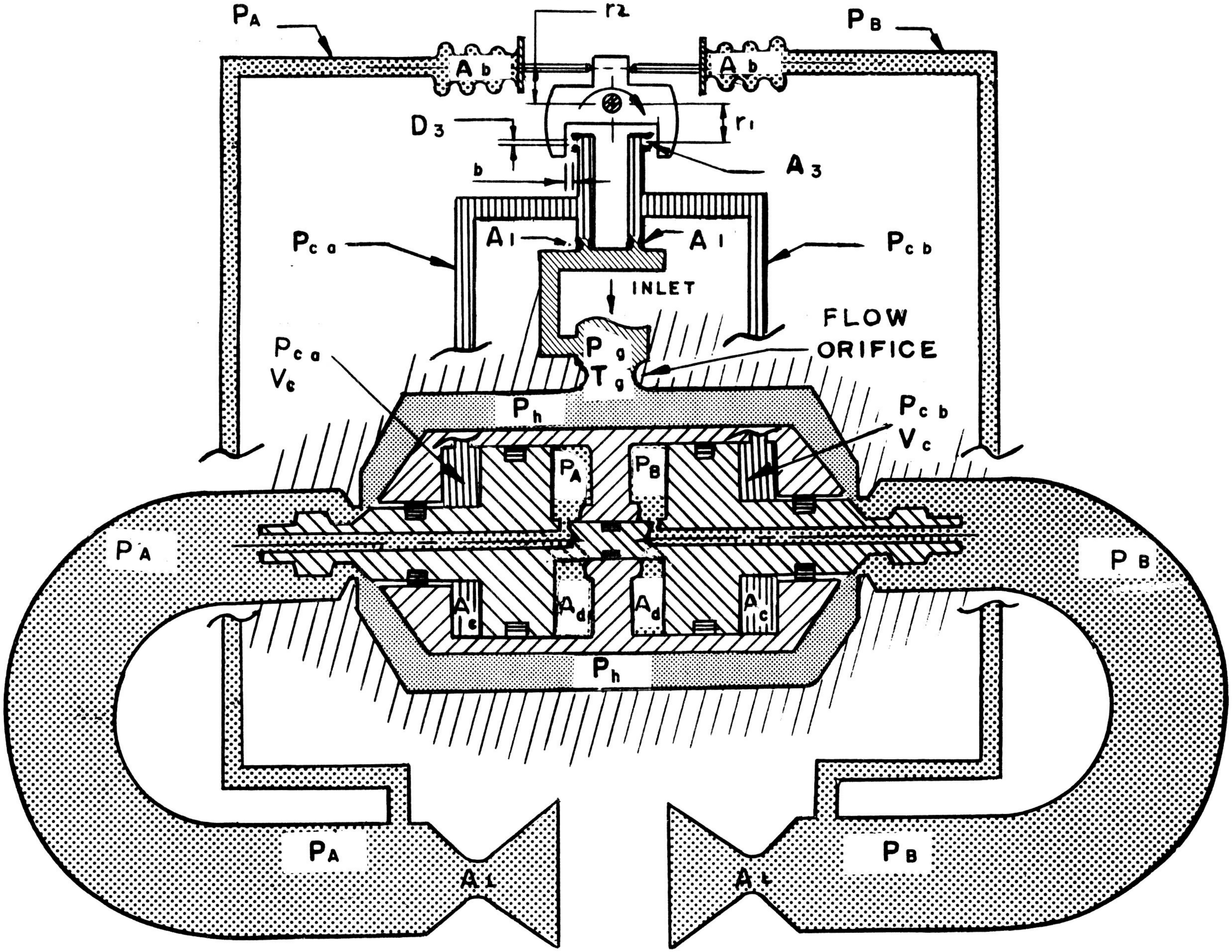
(1) Experimental and published thermodynamic and ballistic properties of Olin-Mathieson OMAX 453D solid propellant.

(2) The combustion gases obey the ideal gas laws.

(3) The mechanism is a constant mass flow device which is supplied by a constant mass flow generator.

(4) A thermal loss of 75°F (based on previous experimental data) is realized from input to output.

(5) All metering and control orifice areas remain unchanged due to thermal effects.



TWO STAGE PNEUMATIC SITVC VALVE

FIGURE – 30

Parameter	2000°F Gas	Gaseous N_2
P_g	2000 psi	2000 psi
P_{ca}	800 to 1950 psi	800 to 1950 psi
P_{cb}	800 to 1950 psi	800 to 1950 psi
P_h	1000 to 1100 psi	1000 to 1100 psi
P_A	0 to 600 psi	0 to 600 psi
P_B	0 to 600 psi	0 to 600 psi
$\dot{\omega}_g$	0.704 lb/sec	1.93 lb/sec
$\dot{\omega}_s$	0.023 lb/sec	0.0625 lb/sec
$\dot{\omega}_A$	0 to .681 lb/sec	0 to 1.85 lb/sec
$\dot{\omega}_B$	0 to .681 lb/sec	0 to 1.85 lb/sec
K	1.279	1.4
R	80.3 ft/°R	55 ft/°R
M	19.25	28
T_g	2000°F	2000°F
T_o	1925°F	1925°F
C_1	0.413 °R/sec	0.522 °R/sec

SYSTEM DESIGN PARAMETERS

FIGURE 31

Basic Flow Equations

The basic equation for sonic pneumatic flow through an orifice is:

$$\dot{W} = C_1 \, C_d \, A \, \frac{P_u}{\sqrt{T_u}}$$

where:

$$C_1 = \sqrt{\frac{KG}{R\left(\frac{K+1}{2}\right)^{\left(\frac{K+1}{K-1}\right)}}} = \text{sonic flow coefficient}$$

C_1 = .413 $\sqrt{°R}$/sec for OMAX 453D

C_1 = .522 $\sqrt{°R}$/sec for Gaseous N_2

The above equation is true only for pneumatic flow in the sonic flow regime. Determination of the sonic flow criteria can be calculated from the critical pressure ratio:

$$\left(\frac{P_d}{P_u}\right)_{Critical} = \left(\frac{2}{K+1}\right)^{\left(\frac{K}{K+1}\right)} = .548 \text{ for OMAX 453D}$$

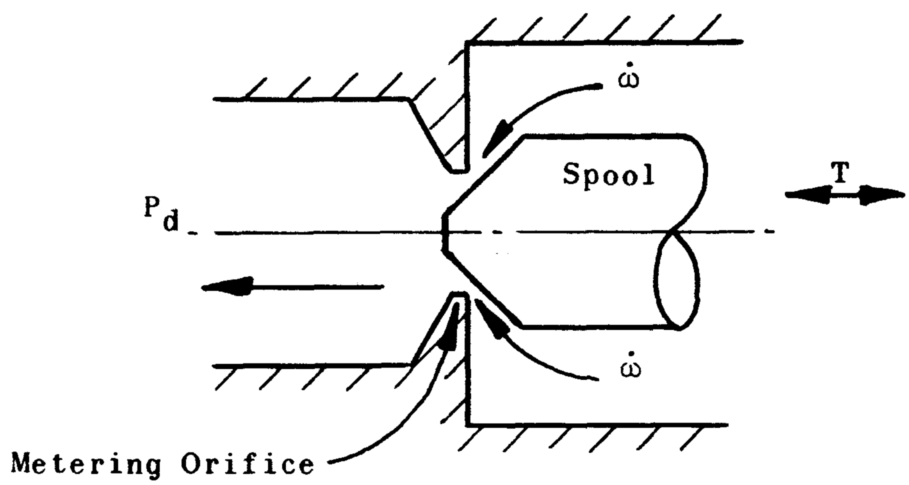
For critical pressure ratios of less than 0.548 then:

$$\frac{\dot{W}_{actual}}{\dot{W}_{max}} = 1.0$$

or pneumatic flow for this condition is independent of downstream pressure and is only a function of upstream pressure.

Since all orifices were basically designed for the sonic flow regime, sub-sonic flow equations were not used in the steady state analysis. The servo orifices which could experience sub-sonic flow in the "off-null" position were included as a non-linear function in the analog computer analysis.

The main spool-metering orifice configuration design was a 90° included angle spool tip and a sharp edge metering orifice.



By using the basic flow equation and solving for effective metering area, then the valve stroke can be determined from the following equation:

$$T\pi \sin\theta \left[D - T^2\pi \sin^2\theta \cos\theta \right] = A_e$$

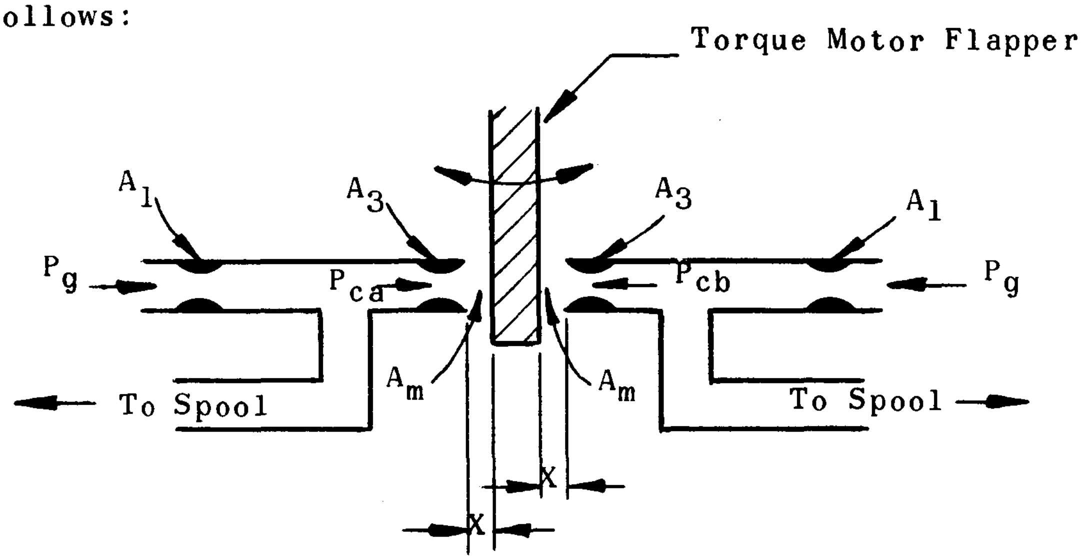
where: T = valve stroke - inches

θ = 1/2 angle of spool tip

D = metering orifice diameter

A_e = effective metering area

The servo design equations and simplified servo schematic are as follows:



An $\alpha \left(\frac{A_m}{A_1}\right)$ ratio of 1.5 was selected for the servo design.

The upstream orifices are sized based on the servo capacity desired in the sonic flow regime.

Thus:

$$A_m = \alpha A_1$$

also, $$A_m = \pi D_3 X$$

An $\frac{A_3}{A_m}$ design ratio of 3.0 minimum was selected to prevent an interaction of metering by A_3.

$$\therefore \quad X = \frac{A_m}{\pi D_3}$$

APPENDIX B

ANALOG COMPUTER ANALYSIS AND STUDY

The analog computer study was conducted in two phases. Phase I was directed specifically to optimization of the dynamics, stability criteria, and parameter sizes for the 0.7 lb/sec flightweight two-stage pneumatic pressure feedback valve. Phase I is discussed in Section 3.1. Phase II was conducted to investigate the effect of major parameter variations on two-stage valve performance. The parameter variations encompassed, (1) 2000°F and 5500°F operating supply temperatures, (2) 1500, 2000, and 2700 psi generator supply pressures, and (3) 0.58, 2.0, and 5.0 lb/sec mass flows.

Since the mathematical logic is identical for the design concept, the selected design parameters and pertinent values were applied and scaled in the analog study.

The analog analysis was conducted using an EAI Pace 221 R and 231 R analog computer. The developed system dynamic equations which mathematically describes the system logic are as follows: [*1]

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*1 See Figure 30 in Appendix A.

TWO-STAGE PNEUMATIC SERVO VALVE SYSTEM

EQUATIONS USED IN ANALOG COMPUTER SIMULATION

1) Main Spool Force Equation:

$$A_c\ (P_{ca}-P_{cb}) = \dot{M}\ d^2T/dt^2 + A_d\ (P_A-P_B) + C\left(\frac{dT/dt}{|dT/dt|}\right) + F_s$$

$$-\dot{M}\ d^2T/dt^2 = -A_c\ (P_{ca}-P_{cb}) + A_d\ (P_A-P_B) + C\left(\frac{dT/dt}{|dT/dt|}\right) + F_s$$

Equation Scaled:

$$-\left(\frac{d^2T/dt^2}{10^6}\right) = -\frac{A_c(3000)}{\dot{M}\cdot 10^6}\left(\frac{P_{ca}-P_{cb}}{3000}\right) + \frac{A_d(1000)}{\dot{M}\cdot 10^6}\left(\frac{P_A-P_B}{1000}\right)$$

$$+\frac{C}{\dot{M}\cdot 10^6}\left(\frac{dT/dt}{|dT/dt|}\right) + \frac{F_s}{\dot{M}\cdot 10^6}$$

2) Flapper Torque Equation:

$$T_i = A_3\ r_1\ (P_{ca}-P_{cb}) + K_r\theta_1 + \delta\dot{\theta}_1 + A_b\ r_2\ (P_A-P_B) + I^{d^2\theta_1/dt^2}$$

$$+\Delta\ \frac{d\theta_1}{dt}$$

$$-I^{d^2\theta_1/dt^2} = -T_i + A_3\ r_1\ (P_{ca}-P_{cb}) + K_r\theta_1 + \delta\dot{\theta}_1 + A_b\ r_2\ (P_A-P_B)$$

$$+\Delta\ \frac{d\theta_1}{dt}$$

Equation scaled:

$$-\left(\frac{d^2\theta_1/dt^2}{2\cdot 10^5}\right) = -\frac{(T_m)}{I\ 2\cdot 10^5}\left(\frac{T_i}{T_m}\right) + \frac{A_3\ r_1(3000)}{I\ 2\cdot 10^5}\left(\frac{P_{ca}-P_{cb}}{3000}\right)$$

$$+\frac{K_r(\theta_{1m})}{I.2.10^5}\left(\frac{\theta_1}{\theta_{1m}}\right) + \frac{A_b\ r_2 x(1000)}{I\cdot 2.10^5}\left(\frac{P_A-P_B}{1000}\right) + \frac{\Delta\ 20}{I\cdot 2\cdot 10^5}\left(\frac{d\theta_1/dt}{20}\right)$$

3 & 4 First Stage Flow Equations

$$3)\quad \frac{C_1\ C_d\ A_1\ P_g\ f(P_{ca})}{\sqrt{T_g}} \quad \frac{C_1\ C_d\ \pi\ D_3}{\sqrt{T_g}}(X-r_1\theta_1)\ P_{ca} + \frac{V_{cm}}{KRT_g}\ \frac{dP_{ca}}{dt}$$

$$+ \frac{A_c\ P_{ca}}{RT_g}\ \frac{dT}{dt} + \frac{C_1\ C_d\ A_L}{\sqrt{T_g}}\ (P_{ca})$$

$$4)\quad \frac{C_1\ C_d\ A_1\ P_g}{\sqrt{T_g}}\ f\ (P_{cb}) = \frac{C_1\ C_d\ \pi\ D_3\ (X+r_1\theta_1)}{\sqrt{T_g}}\ P_{cb} + \frac{V_m}{KRT_g}\ \frac{dP_{cb}}{dt}$$

$$- \frac{A_c\ P_{cb}}{RT_g}\ \frac{dT}{dt} + \frac{C_1\ C_d\ A_L}{\sqrt{T_g}}(P_{cb})$$

$$3)\quad \frac{V_m}{KRT_g}\ \frac{dP_{ca}}{dt} = \frac{C_1\ C_d\ A_3\ P_g}{\sqrt{T_g}}\ f\ (P_{ca}) - \frac{C_1\ C_d\ \pi\ D_3\ (X-r_1\theta_1)}{\sqrt{T_g}}\ P_{ca}$$

$$- \frac{A_c\ \ P_{ca}}{RT_g}\ \frac{dT}{dt} - \frac{C_1\ C_d\ A_L}{\sqrt{T_g}}\ (P_{ca})$$

$$4)\quad \frac{V_m}{KRT_g}\ \frac{dP_{cb}}{dt} = \frac{C_1\ C_d\ A_1\ P_g}{\sqrt{T_g}}\ f\ (P_{cb}) - \frac{C_1\ C_d\ \pi\ D_3\ (X+r_1\theta_1)}{\sqrt{T_g}}\ P_{cb}$$

$$+ \frac{A_c\ \ P_{cb}}{RT_g}\ \frac{dT}{dt} - \frac{C_1\ C_d\ A_L}{\sqrt{T_g}}\ (P_{cb})$$

$$A_L = \frac{A_1\ \ P_g}{P_{ca}}\ (\eta\%)$$

Equations Scaled:

3) $$\left(\frac{dPca/dt}{10^6}\right) = \left(\frac{KRT_g}{V_m \cdot 10^6}\right) \frac{C_1\ C_d\ A_1\ P_g}{\sqrt{T_g}} (f\ P_{ca}) - \frac{KRT_g}{V_m \cdot 10^6}$$

$$\frac{C_1\ C_d\ \pi\ D_3\ S(3000)}{\sqrt{T_g}} \left(\frac{Pca}{3000}\right) + \left(\frac{KRT_g}{V_m \cdot 10^6}\right) \frac{C_1\ C_d\ \pi\ D_3\ r_1\ (\theta_{1m})(3000)}{\sqrt{T_g}}$$

$$\left(\frac{Pca}{3000}\right)\left(\frac{\theta_1}{\theta_{1m}}\right)$$

$$-\left(\frac{KRT_g}{V_m \cdot 10^6}\right)\frac{A_oL}{RT_g}\ (3000)(100)\left(\frac{P_c}{3000}\right)\left(\frac{dT/dt}{100}\right) - \left(\frac{KRT_g}{V_m \cdot 10^6}\right)\ \frac{C_1C_d}{\sqrt{T_g}}\ \left(\frac{A_3P_g(\eta\%)}{Pca}\right)$$

$$(3000)\left(\frac{P_{ca}}{3000}\right)$$

4) $$\left(\frac{dP_{cb}/dt}{10^6}\right) = \left(\frac{KRT_g}{V_m \cdot 10^6}\right)\frac{C_1\ C_d\ A_1\ P_g}{\sqrt{T_g}}\ (f\ P_{ca}) - \left(\frac{KRT_g}{V_m \cdot 10^6}\right)\frac{C_1\ C_d\ \pi\ D_3\ X(3000)}{\sqrt{T_g}}$$

$$\left(\frac{P_{cb}}{3000}\right) - \frac{KRT_g}{V_m \cdot 10^6}\quad \frac{C_1\ C_d\ \pi\ D_3\ r_1\ (\theta_{1m})\ (3000)}{\sqrt{T_g}}\left(\frac{P_{cb}}{3000}\right)\left(\frac{\theta_1}{\theta_m}\right)$$

$$+\ \frac{KRT_g}{V_m \cdot 10^6}\ \frac{A_L\ (3000)(100)}{RT_g}\left(\frac{P_c}{3000}\right)\left(\frac{dT/dt}{100}\right) - \left(\frac{KRT_g}{V_m \cdot 10^6}\right)$$

$$\frac{C_1\ C_d}{\sqrt{T_g}}\quad \frac{A_1\ P_g\ (\eta\%)}{Pca}\ (3000)\left(\frac{P_{cb}}{3000}\right)$$

Second Stage Flow Equations

$$5)\quad \frac{C_1\ C_c\ P_d}{\sqrt{T_g}} K_a T = \frac{C_1\ C_d\ AP_A}{\sqrt{T_g}} + \frac{V_A}{KRT_g}\ \frac{dP_A}{dt}$$

$$6)\quad \frac{C_1\ C_d\ P_d}{\sqrt{T_g}} (-K_a T) = \frac{C_1\ C_d\ AP_B}{\sqrt{T_g}} + \frac{V_B}{KRT_g}\ \frac{dP_B}{dt}$$

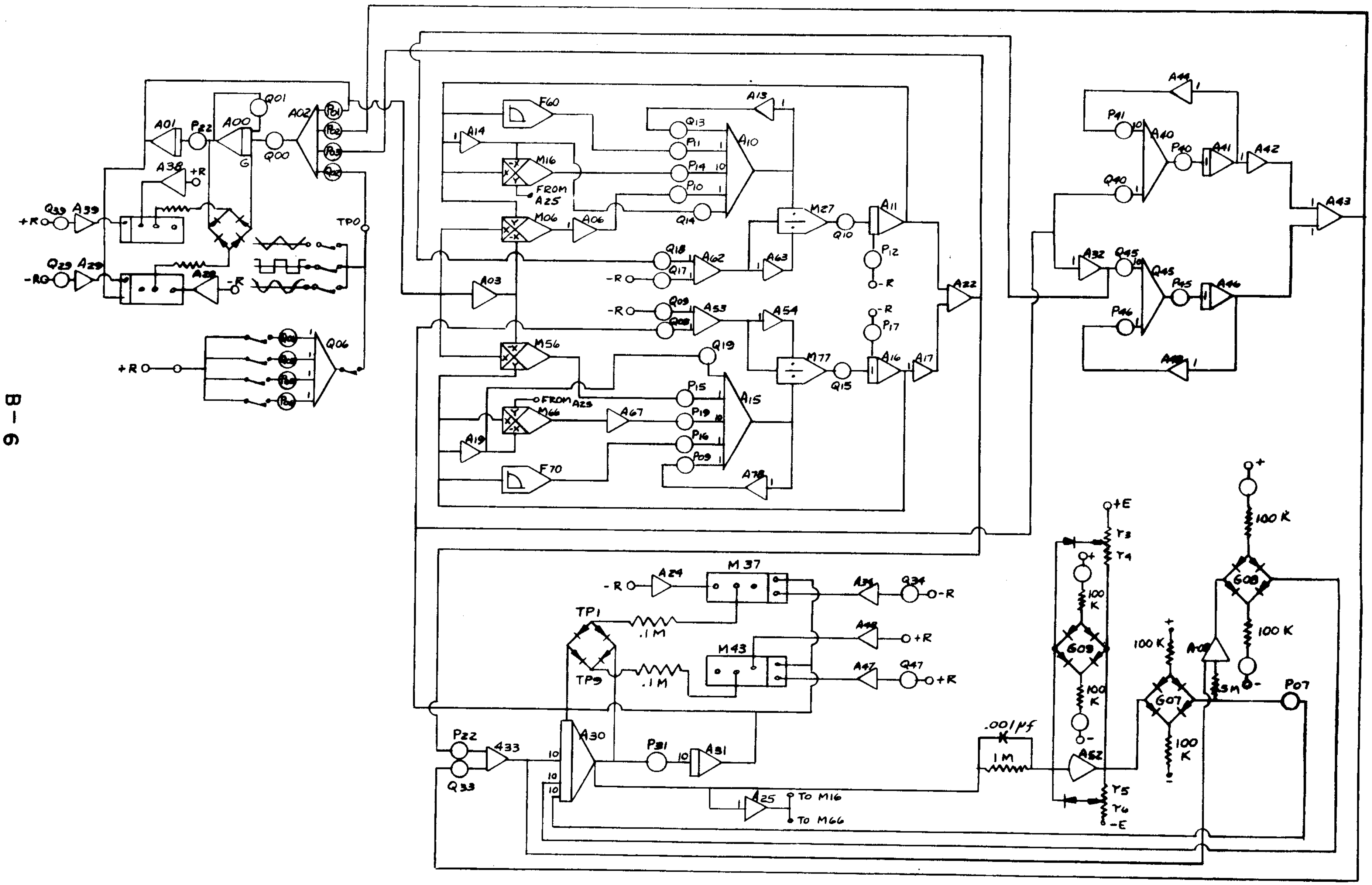
Scaled Equations:

$$5)\quad \left(\frac{dP_A/dt}{10^5}\right) = \left(\frac{KRT_g}{V_A \cdot 10^5}\right) \frac{C_1\ C_d\ P_A\ K_a\ T_m}{\sqrt{T_g}} \left(\frac{T}{T_m}\right) = \frac{C_1\ C_d\ A(1000)}{\sqrt{T_g}} \left(\frac{P_A}{1000}\right)$$

$$6)\quad \left(\frac{dP_B/dt}{10^5}\right) = -\left(\frac{KRT_g}{V_B\ 10^5}\right) \frac{C_1\ C_d\ P_B\ K_a\ T_m}{\sqrt{T_g}} \left(\frac{T}{T_m}\right) = \frac{C_1\ C_d\ A(1000)}{\sqrt{T_g}} \left(\frac{P_A}{1000}\right)$$

The mathematical description of the system was scaled and translated to computer language. Figure 32 presents the computer block diagram for analog application of the system equations.

For evaluation of the effect of major parameter variations on valve concept performance, seven (7) valves were sized to enable incorporation of actual parameters and pertinent sizes in the analog program. These valves were designated as H-20L, M-20L, etc., only as code means of recognizing the valve parameters and sizes. For example, H-20L denotes high pressure (2700 psi) - 2000°F system, - low flow



ANALOG COMPUTER BLOCK DIAGRAM

FIGURE - 32

(0.704 lb/sec). These codes and tabulation of all pertinent information is shown on Figure 33.

The analog results of these studies are exhibited in Figures 34 through 39.

FIGURE 33

TABULATION OF PARAMETERS AND SIZES FOR

COMPUTER SIMULATION OF PRESS-TEMP-FLOW VARIATIONS

TWO-STAGE HIGH TEMP PNEUMATIC TVC VALVE

Symbol	Description or Parameter	Units	H-20L	M-20L	M55/20L	M55L	M55M	M55H	H55L
P_g	Generator pressure	psia	2015	1515	1515	1515	1515	1515	2015
T_g	Generator output temperature	°F	2000	2000	5500	5500	5500	5500	5500
$\dot{\omega}_g$	Generator mass flow output	lb/sec	0.704	0.704	0.704	0.704	2.0	5.0	0.704
Ps	Servo generator pressure	psia	----	----	1515	----	----	----	----
Tgs	Servo generator output temperature	°F	----	----	2000	----	----	----	----
$\dot{\omega}_s$	Servo generator mass flow output	lb/sec	----	----	0.028	----	----	----	----
--	Main flow propellant (1) OMAX453D ammonia nitrate (2) Arcite 373 ammonium perchlorate	----	OMAX453D	OMAX453D	Arcite 373	Arcite 373	Arcite 373	Arcite 373	Arcite 373
--	Servo flow propellant	----	OMAX453D	OMAX453D	OMAX453D	Arcite 373	Arcite 373	Arcite 373	Arcite 373
R	Propellant gas constant	Ft/°R	80.3	80.3	52.38/80.3 Servo	52.38	52.38	52.38	52.38
K	Ratio of specific heats (Cp/Cv)	----	1.28	1.28	1.185/1.28 Servo	1.185	1.185	1.185	1.185
M	Gas molecular weight	lbs	19.25	19.25	29.5/19.25 Servo	29.5	29.5	29.5	29.5
Isp	Propellant specific impulse (1000 - 14.7) psia	sec	165	165	257.7/165 Servo	257.7	257.7	257.7	257.7
C_1	Orifice flow coefficient (sonic)	$\sqrt{°R}$/sec	0.413	0.413	0.513/0.413 Servo	0.513	0.513	0.513	0.513
C_2	Orifice Flow coefficient (subsonic)	$\sqrt{°R}$/sec	1.902	1.902	2.803/1.902 Servo	2.803	2.803	2.803	2.803
Pg	Valve inlet and servo inlet pressure	psia	2015	1515	1515	1515	1515	1515	2015
P_d/P_u	Sonic flow critical pressure ratio	----	0.552	0.552	0.566/0.552 Servo	0.566	0.566	0.566	0.566
P_h	Valve case press(downstream of inlet sonic mass flow orifice)	psia	1000	765	765	765	765	765	1000
P_{An} or P_{on}	Valve output pressure - null	psia	315	202	202	202	202	202	315
P_A or P_B	Valve output pressure min-max range	psia	0 to 615	0 to 390	0 to 390	0 to 390	0 to 390	0 to 390	0 to 615
P_{CN}	Valve servo control pressure - null	psia	1100	865	865	865	865	865	1100

FIGURE 33

Symbol	Description or Parameter	Units	H-20L	M-20L	M55/20L	M55L	M55M	M55H	H55L
Δ Pc	Valve servo control pressure - min-max range	psia	500-2000	350-1515	350-1515	350-1515	350-1515	350-1515	500-2000
P_{Ad} or P_{Bd}	Valve output pressure feedback - null	psia	315	202	202	202	202	202	315
P_{Ad} or P_{Cd}	Valve output pressure feedback - min-max range	psia	0 to 615	0 to 390	0 to 390	0 to 390	0 to 390	0 to 390	0 to 615
T_g	Valve inlet gas temperature (estimated)	°F	1965	1965	5490/1965 Servo	5450	5450	5450	5450
T_A or T_B	Valve discharge gas temperature	°F	1850	1850	5400/1900 Servo	5400	5400	5400	5400
I	Torque motor sum current (push-pull circuit)	M-amps	50	50	50	50	50	50	50
T_i	Torque motor output torque (May)	lb/in	4.0	5.0	5.0	5.0	40	80	4.0
W	Torque motor power consumption	watts	5.5	6.0	6.0	6.0	----	----	5.5
Im	Torque motor yoke moment of inertia	in/lb sec^2	$2.1x10^{-5}$	$2.1x10^{-5}$	$2.1x10^{-5}$	$2.1x10^{-5}$	$5x10^{-5}$	$9.5x10^{-5}$	$2.1x10^{-5}$
$\dot{\omega}$3	Servo flow capacity - %	%	3.27	3.27	3.16	3.27	4.0	4.0	3.27
$\dot{\omega}$s	Servo flow capacity	lb/sec	0.023	0.023	0.023	0.023	0.08	0.20	0.023
$\dot{\omega}$v	Net main stage output flow	lb/sec	0.681	0.681	0.704	0.681	1.92	4.8	0.681
X	Servo yoke gap @ null	Radians	±0.0180	±0.0264	±0.0264	±0.0314	±0.016	±0.0372	±0.020
T	Servo yoke stroke (from null)	inches	±0.00452	±0.0066	±0.0066	±0.00786	±0.0126	±0.0231	±0.005
r_1	Servo yoke to torque motor C Radius	inches	0.250	0.250	0.250	0.250	0.375	0.625	0.250
A1	Area of upstream servo orifice	$inch^2$	0.000548	0.000964	0.000964	0.00116	0.00404	0.0101	0.000627
D_1	Dia.of upstream servo orifice	inches	0.0270	0.035	0.035	0.0385	0.072	0.1135	0.028
A_3	Area of downstream servo orifice	$inch^2$	0.00458	0.00675	0.00675	0.00695	0.0323	0.0606	0.00503
D_3	Dia. of downstream servo orifice	inches	0.0764	0.093	0.093	0.094	0.203	0.278	0.080
A_{MS}	Effective area of servo metering orifice @ null	inches	0.00108	0.00193	0.00193	0.00232	0.00809	0.0202	0.00126
C_{ds}	Discharge coefficient - servo orifices	----	0.98	0.98	0.98	0.98	0.98	0.98	0.98
α	servo metering orifice/upstream orifice area ratio	----	2.0	2.0	2.0	2.0	2.0	2.0	2.0
A_{ds}/A_{ms}	$\frac{\text{Area of downstream orifice}}{\text{Area of metering orifice}}$ Ratio	----	4.25	3.5	3.5	3.0	4.0	3.0	4.0
A_{TC}	Area of servo control transmission line	$inch^2$	0.0122	0.0202	0.0202	0.0209	0.097	0.1818	0.0151
D_{TC}	Dia. of servo control transmission line	inches	0.125	0.160	0.160	0.163	0.352	0.480	0.139
V_c	Compressibility - volume servo - one side (inch - peppet piston volume) at null	$inch^3$	0.092	0.100	0.110	0.110	0.161	0.22	0.092

FIGURE 33

Symbol	Description or Parameter	Units	H-20L	M-20L	M55/20L	M55L	M55M	M55H	H55L
r_2	Radius of TM to feedback bellows	inches	0.130	0.130	0.130	0.130	0.250	0.375	0.130
K_B	Feedback bellows spring rate	lb/in	40	40	40	40	150	150	40
F_B	Feedback bellows preload	lbs	0.16	0.16	0.16	0.16	1.2	2.1	0.16
A_b	Feedback bellows effective area	$inch^2$	0.023	0.023	0.023	0.023	0.173	0.489	0.023
$A_B r_{FB}$	Feedback torque constant	$inch^3$	0.00355	0.00355	0.00355	0.00355	0.0432	0.183	0.00355
V_B	Feedback bellows volume @ null	$inch^3$	0.062	0.062	0.062	0.062	0.218	0.578	0.062
V_{FT}	Volume of feedback transmission line (one side)	$inch^3$	0.134	0.134	0.134	0.134	0.648	1.76	0.134
X_B	Bellows stroke from null ±	inch	0.00234	0.00343	0.00343	0.00405	0.0084	0.0138	0.0026
A_P	Output spool metering area (total)	$inch^2$	0.0826	0.927	0.117	0.1125	0.386	0.962	0.103
T	Output spool stroke null ±	inch	±0.055	±0.066	±0.075	±0.074	±0.125	±0.170	±0.089
A_{mo}	Spool metering orifice area (net)	$inch^2$	0.072	0.0962	0.0962	0.0962	0.441	1.4849	0.072
D_{mo}	Spool metering orifice diameter	inch	0.304	0.385	0.385	0.385	0.750	1.375	0.304
A_A or A_B	Output throat area (nozzle A or B)	$inch^2$	0.132	0.214	0.273	0.265	0.729	1.82	0.168
D_A or D_B	Output throat dia.(nozzle A or B)	inch	0.410	0.522	0.590	0.581	0.964	1.523	0.463
A_o	Area of valve inlet sonic orifice	$inch^2$	0.0407	0.0552	0.0705	0.068	0.191	0.476	0.513
D_o	Diameter of valve inlet sonic orifice	inch	0.228	0.265	0.300	0.295	0.493	0.779	0.256
A_c	Servo control effective area	$inch^2$	0.3167	0.540	0.600	0.600	0.970	1.30	0.3167
D_c	Servo control piston bore dia	inch	0.750	1.035	1.083	1.083	1.497	2.020	0.750
A_d	Pneumatic rate effective area	$inch^2$	0.371	0.77	0.85	0.85	0.616	1.58	0.371
D	Poppet bore diameter	inch	0.400	0.804	0.934	0.834	1.202	1.738	0.400
A_{cs}	Area of center shaft bore	$inch^2$	0.071	0.071	0.071	0.071	0.071	0.071	0.071
D_{cs}	Dia of center shaft bore	inch	0.300	0.300	0.300	0.300	0.300	0.300	0.300
A_{pt}	Area of pneumatic rate transmission line	$inch^2$	0.0069	0.012	0.012	0.012	0.027	0.049	0.0069
D_{pt}	Dia of pneumatic rate transmission line	inch	0.093	0.125	0.125	0.125	0.187	0.250	0.093
K_s	Equivalent pneumatic spring rate	lb/in	2930	2860	2800	2840	2710	2680	1920

FIGURE 33

Symbol	Description or Parameter	Units	H-20L	M-20L	M55/20L	M55L	M55M	M55H	H55L
K_R	Mechanical return spring rate	lb/in	50	50	50	50	100	200	50
K_V	Net valve spring rate	lb/in	2840	3590	3250	3290	3240	3285	2150
M	Mass of spool	$lb/sec^2/in$	2.867×10^{-4}	7.2×10^{-4}	7.2×10^{-4}	7.2×10^{-4}	1.82×10^{-3}	3.10×10^{-3}	2.86×10^{-4}
C_s	Poppet coulomb stiction (break-away)	lbs	60	86	86	86	172	243	60
C_c	Poppet coulomb friction (dynamic)	lbs	50	72	72	72	144	203	50
$\dot{\omega}_A$	Poppet metered flow - null "A"	lb/sec	0.3405	0.3405	0.352	0.3405	0.96	2.4	0.3405
$\dot{\omega}_B$	Poppet metered flow - null "B"	lb/sec	0.3405	0.3405	0.352	0.3405	0.96	2.4	0.3405
V_P	Poppet slew rate	in/sec	12.6	12.6	14.1	14.1	15.7	21.3	16.7
F	Frequency response design	cps	30	30	30	30	20	20	30
V_A	Volume of "A" output transmission line	$inch^3$	8.05	8.05	8.05	8.05	20	53	8.05
V_B	Volume of "B" output transmission line	$inch^3$	8.05	8.05	8.05	8.05	20	53	8.05
N_A	Volume of "A" output nozzle	$inch^3$	1.37	1.37	1.37	1.37	2.5	5.0	1.37
N_B	Volume of "B" output nozzle	$inch^3$	1.37	1.37	1.37	1.37	2.5	5.0	1.37

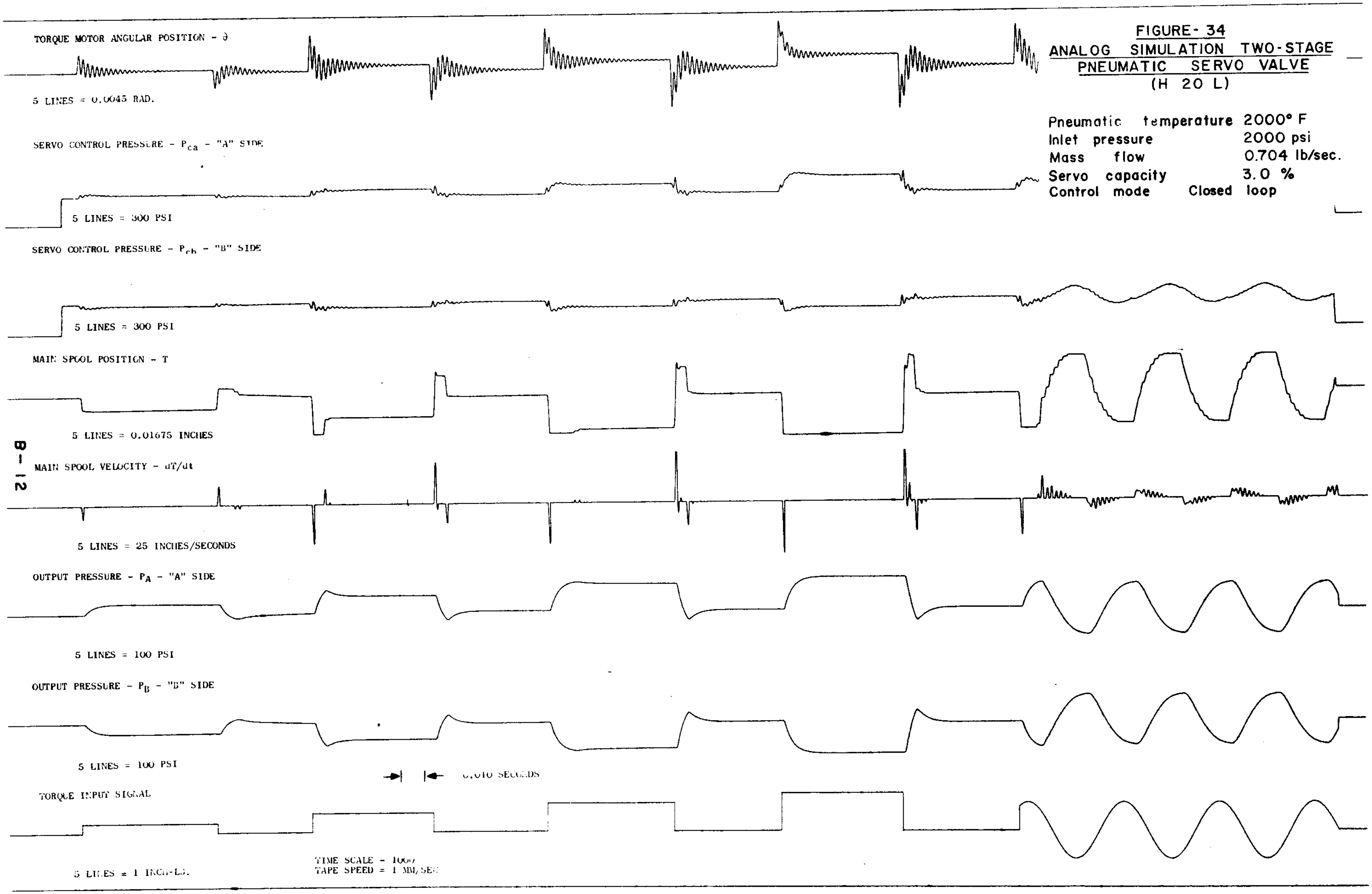


FIGURE- 34
ANALOG SIMULATION TWO-STAGE
PNEUMATIC SERVO VALVE
(H 20 L)

Pneumatic temperature 2000° F
Inlet pressure 2000 psi
Mass flow 0.704 lb/sec.
Servo capacity 3.0 %
Control mode Closed loop

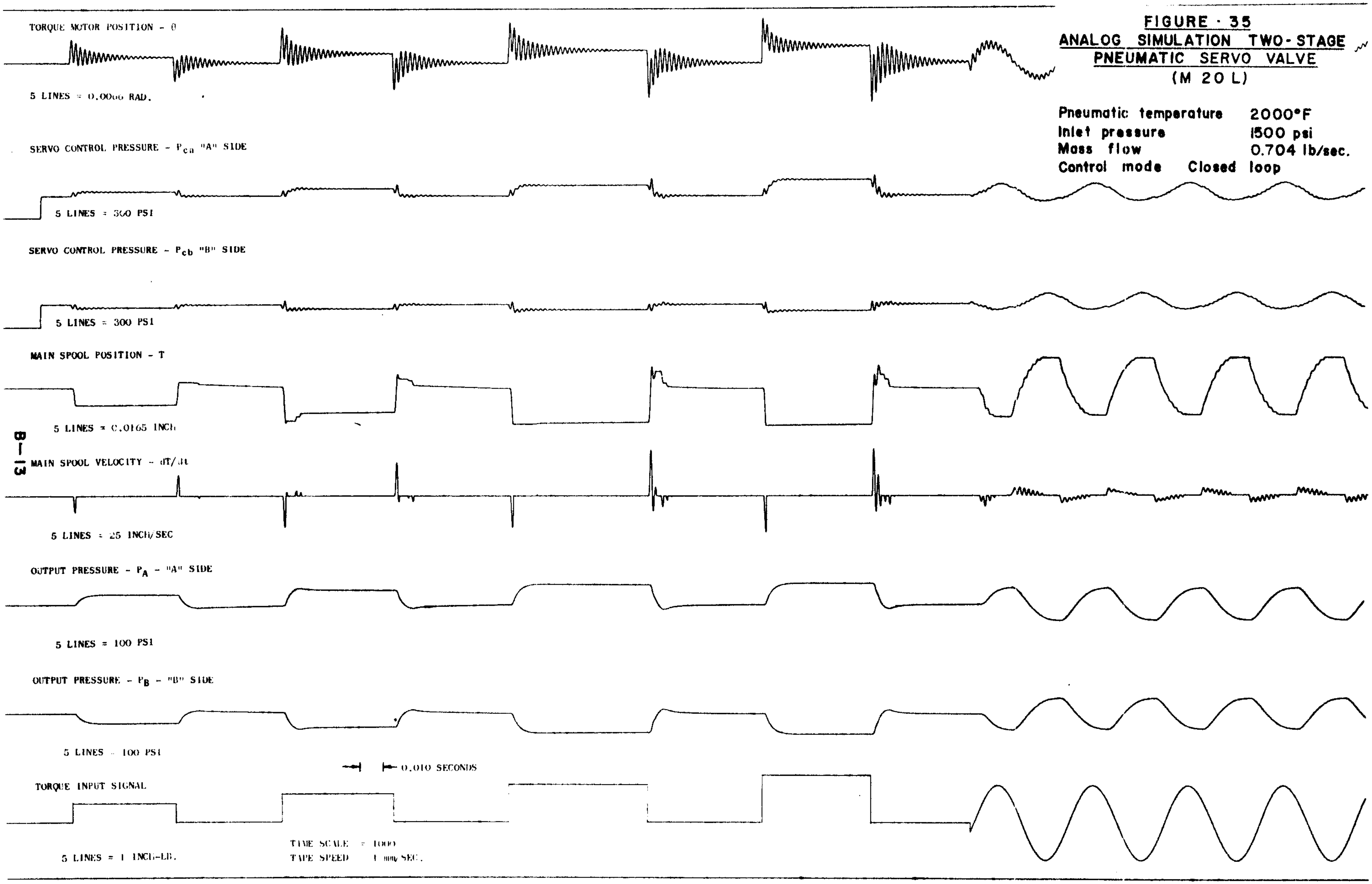


FIGURE · 35
ANALOG SIMULATION TWO-STAGE PNEUMATIC SERVO VALVE
(M 20 L)

Pneumatic temperature	2000°F
Inlet pressure	1500 psi
Mass flow	0.704 lb/sec.
Control mode	Closed loop

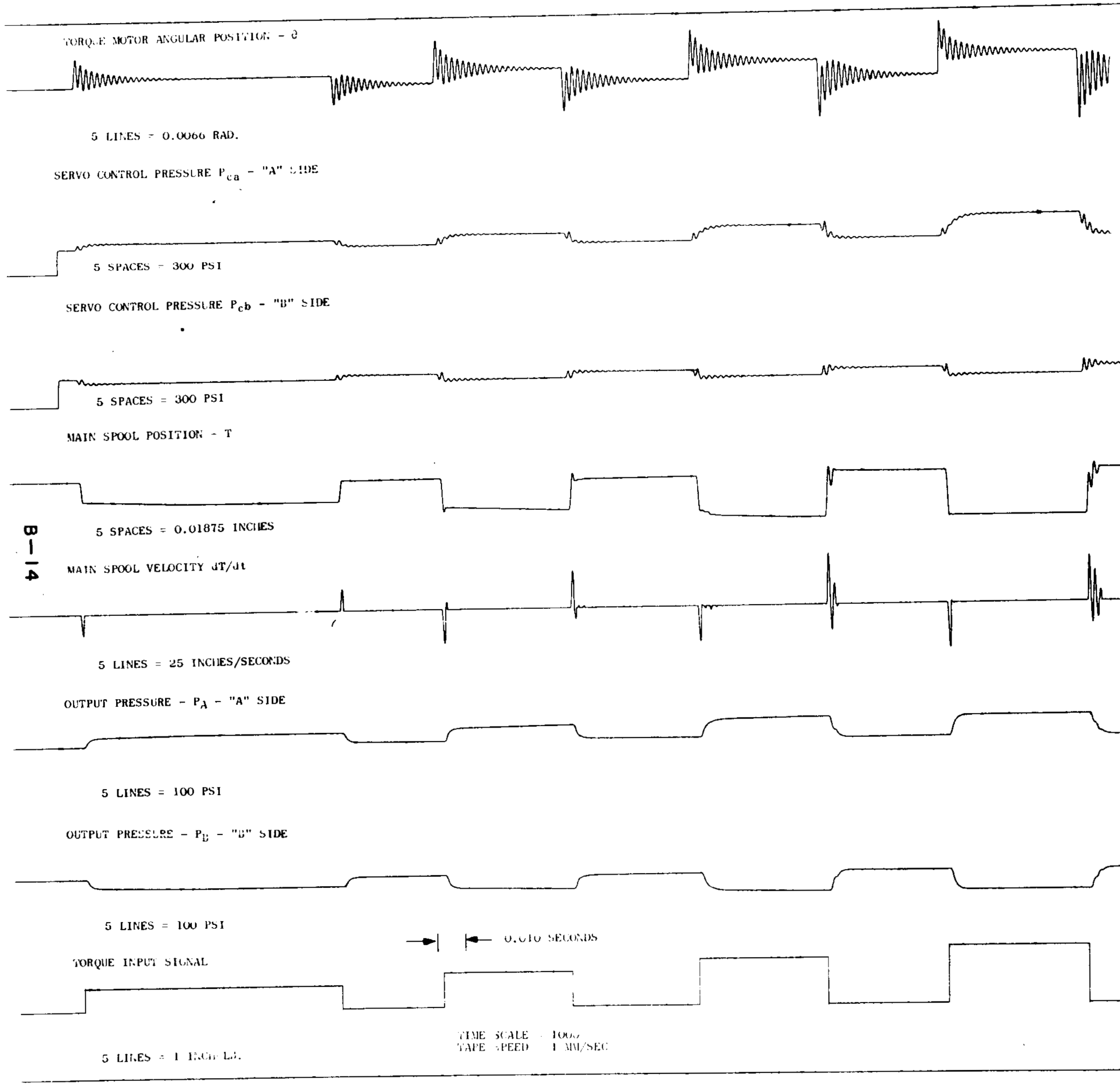


FIGURE - 36
ANALOG SIMULATION TWO-STAGE
PNEUMATIC SERVO VALVE
(M 55/20 L)

Pneumatic temperature	5500°F
Servo inlet temperature	2000°F
Inlet pressure	1500 psi
Mass flow	0.704 lb/sec.
Control mode	Closed loop

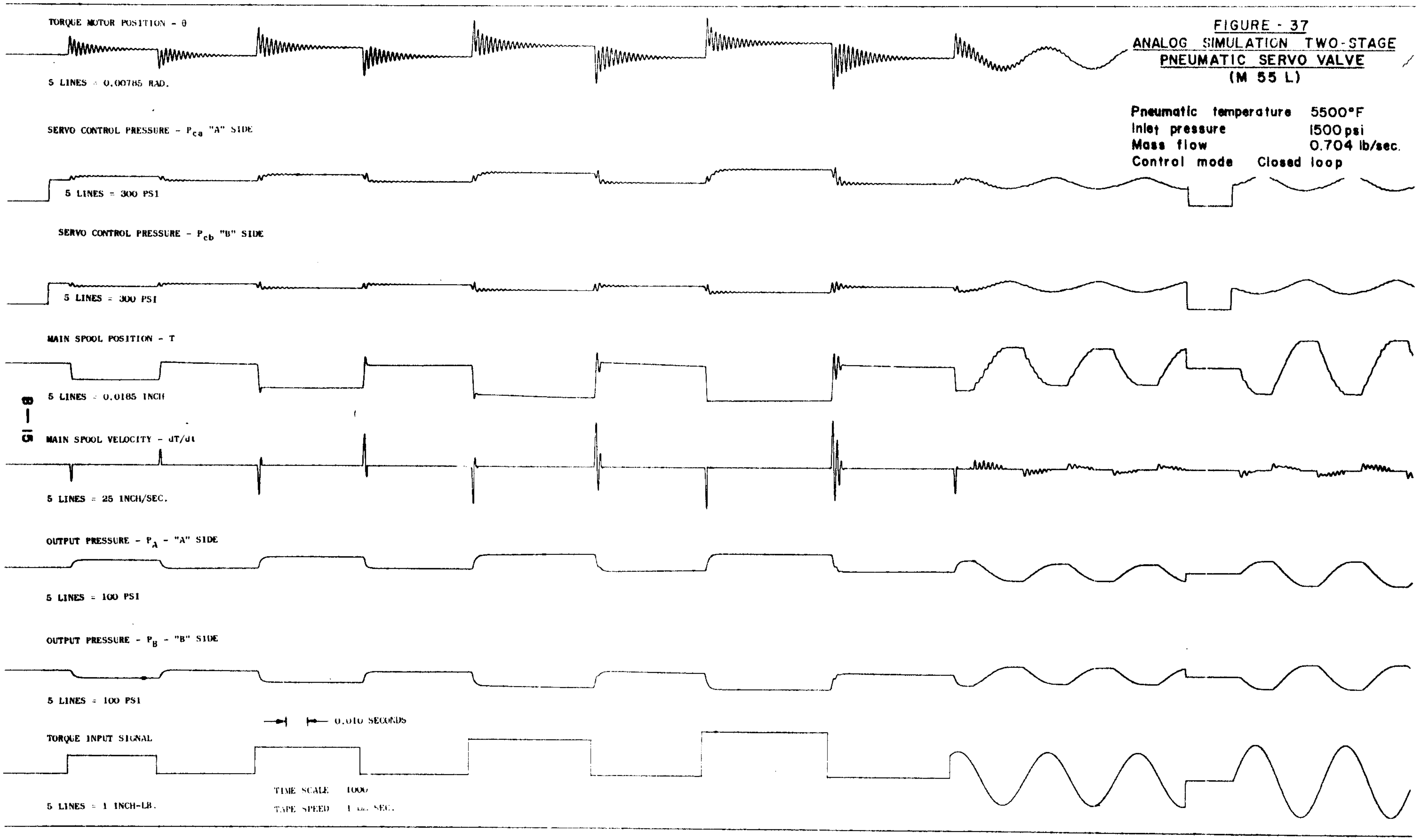


FIGURE - 37
ANALOG SIMULATION TWO-STAGE
PNEUMATIC SERVO VALVE
(M 55 L)

Pneumatic temperature 5500°F
Inlet pressure 1500 psi
Mass flow 0.704 lb/sec.
Control mode Closed loop

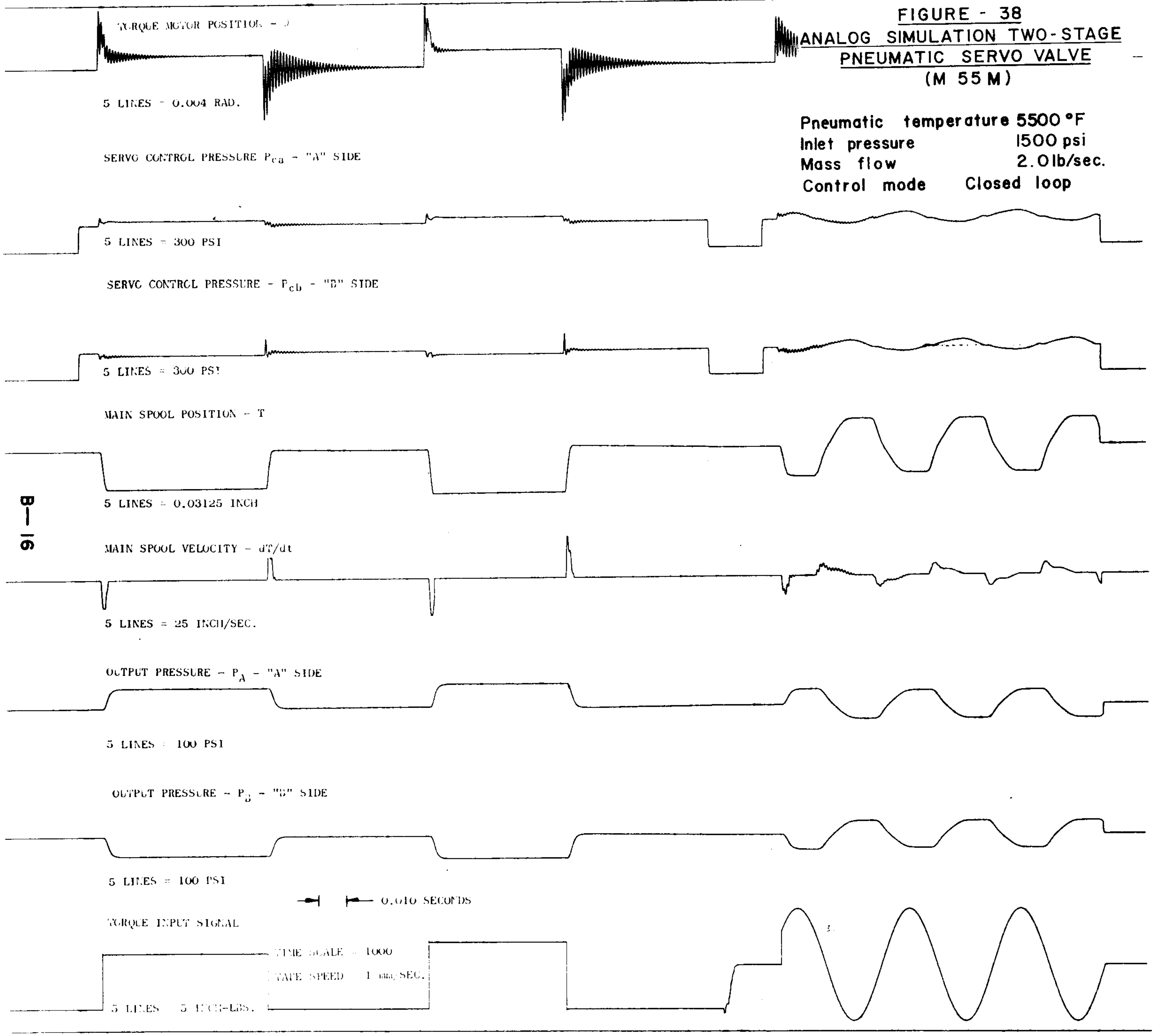


FIGURE - 38
ANALOG SIMULATION TWO-STAGE
PNEUMATIC SERVO VALVE
(M 55 M)
Pneumatic temperature 5500 °F
Inlet pressure 1500 psi
Mass flow 2.0 lb/sec.
Control mode Closed loop
TORQUE MOTOR POSITION
5 LINES = 0.004 RAD.
SERVO CONTROL PRESSURE P_{ca} - "A" SIDE
5 LINES = 300 PSI
SERVO CONTROL PRESSURE - P_{cb} - "B" SIDE
5 LINES = 300 PSI
MAIN SPOOL POSITION - T
5 LINES = 0.03125 INCH
MAIN SPOOL VELOCITY - dT/dt
5 LINES = 25 INCH/SEC.
OUTPUT PRESSURE - P_A - "A" SIDE
5 LINES = 100 PSI
OUTPUT PRESSURE - P_B - "B" SIDE
5 LINES = 100 PSI
0.010 SECONDS
TORQUE INPUT SIGNAL
TIME SCALE = 1000
TAPE SPEED 1 mm/SEC.
5 LINES = 5 INCH-LBS.

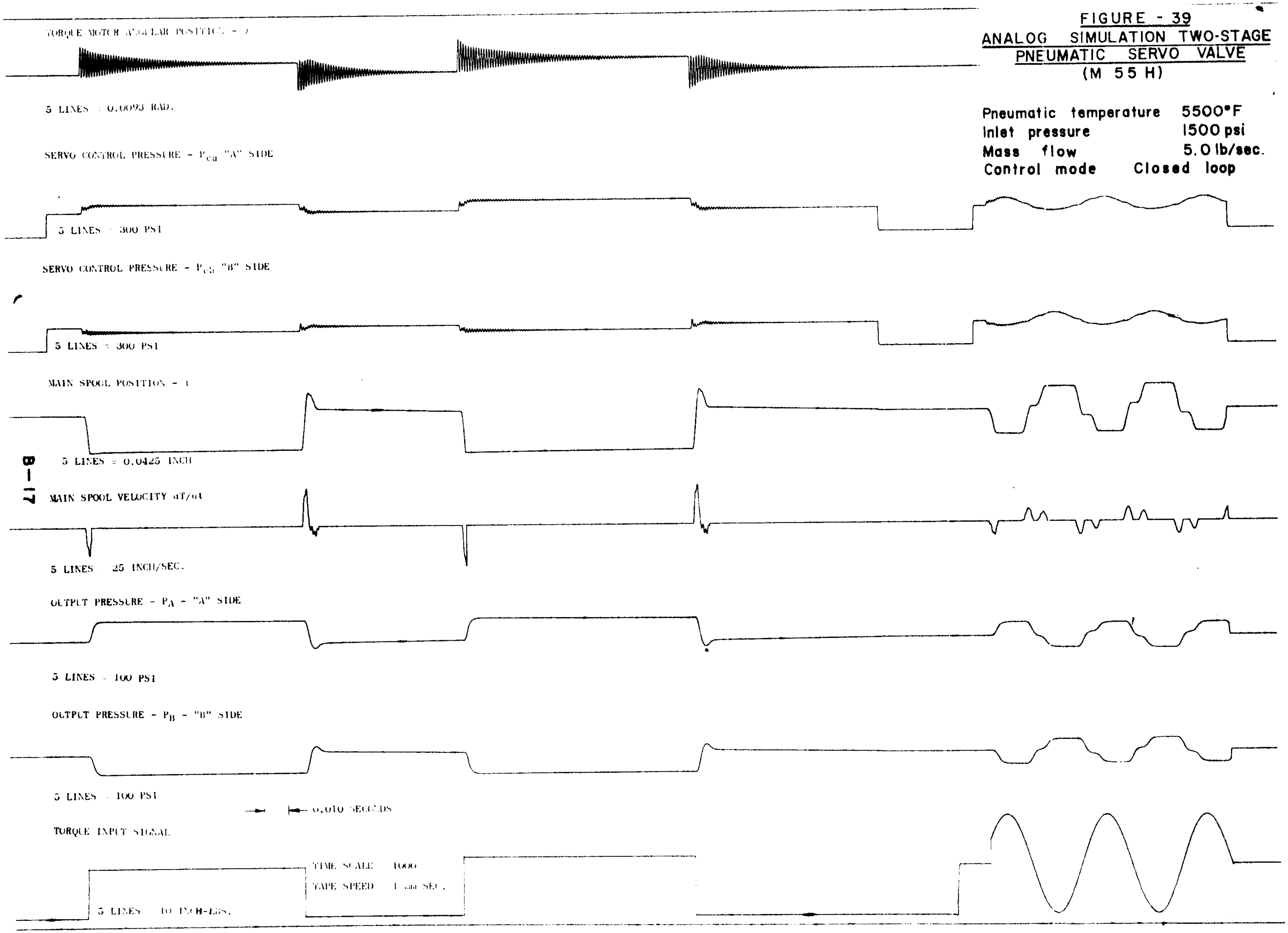


FIGURE - 39

ANALOG SIMULATION TWO-STAGE PNEUMATIC SERVO VALVE (M 55 H)

Pneumatic temperature 5500°F
Inlet pressure 1500 psi
Mass flow 5.0 lb/sec.
Control mode Closed loop

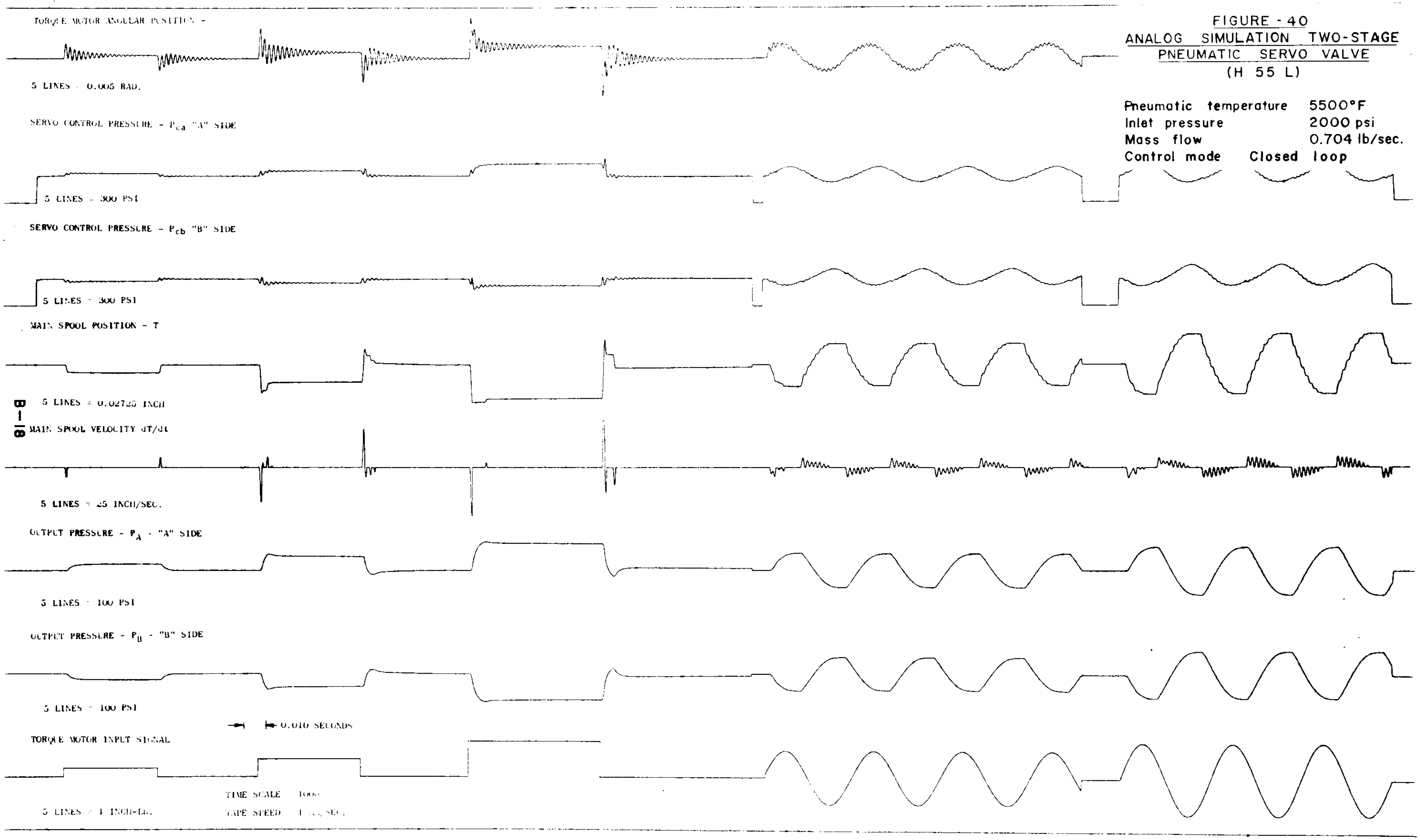


FIGURE - 40
ANALOG SIMULATION TWO-STAGE
PNEUMATIC SERVO VALVE
(H 55 L)

Pneumatic temperature 5500°F
Inlet pressure 2000 psi
Mass flow 0.704 lb/sec.
Control mode Closed loop

APPENDIX C

TWO-STAGE VALVE DEVELOPMENT
TEST EQUIPMENT AND INSTRUMENTATION

(1) Voice of Music Tape-o-Matic 730 Monophonic Tape Recorder, Lab #3414-R

(2) Sanborn 150 - 6 Channel Recorder,
Lab #3346-R - Carrier Preamplifier - Model 1100AS
Lab #3072-C - Carrier Preamplifier - Model 1100AS
Lab #2264-C - AC & DC Preamplifier - Model 1000
Lab #3344-R - Carrier Preamplifier - Model 1100AS
Lab #3345-R - Carrier Preamplifier - Model 1100AS
Lab #3073-C - Carrier Preamplifier - Model 1100AS

(3) Vickers Model 56243-X Power Supply Differential Current, Lab #2774-R

(4) Model 56242-X Power Amplifier, Lab #2574-R

(5) Loop Switch Panel and Gain Control, Lab #2784-R

(6) Hewlett Packard, Palo Alto, California

(7) Model 211A Square Wave Generator, Lab #2052-C

(8) Hewlett Packard, Palo Alto, California

(9) Model 122AR Oscilloscope, Lab #2807-R

(10) Minneapolis Honeywell, Denver, Colorado

(11) Model 1108 - 26700HK Visicorder

(12) Century Electronics and Instruments, Inc., Tulsa, Oklahoma

(13) Model 808 Power Supply and KC Oscillator, Lab #1341

(14) Century Electronics and Instruments, Inc., Tulsa, Oklahoma

(15) Model 408 Recording Oscillograph

(16) Case Press - Teledyne Pressure Transducer, Lab #2698 (5 K psi)

(17) Thermocouple - C/A

(18) P_{cA} - Dynisco, Lab #2522 (10 K psi)

(19) P_{cB} - Dynisco, Lab #2361 (10 K psi)

(20) P_A - Dynisco, Lab #3098 (1000 psi)

(21) P_B - Dynisco, Lab #2119 (1000 psi)

(22) P_{inlet} - Dynisco, Lab #2256 (10 K psi)

(23) P_A - Dynisco, Lab #3194 (1000 psi)

(24) P_B - Dynisco, Lab #3654 (1000 psi)

(25) Hewlett Packard, Palo Alto, California
Model 202A Low Frequency Function Generator, Lab #2757-R

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